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HEAT TRANSMISSION THROUGH BOILER TUBES

BY

HUBER O. CROFT

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ENGINEERING EXPERIMENT STATION

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HEAT TRANSMISSION THROUGH BOILER TUBES

BY

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HEAT TRANSMISSION THROUGH BOILER TUBES

I. Introduction

1. *Object and Scope.*—In the past boiler heat transmission data, which have been obtained under laboratory control methods, have been determined on the "fire-tube" type of apparatus where the boiling phenomena and water circulation on the wet side of the heating surface probably did not duplicate the conditions which exist in an actual power boiler. The objects of this investigation were to obtain heat transmission data under conditions similar to those existing in an actual water-tube boiler and to study the phenomena of water circulation under the same conditions.

2. *General Scheme of Test Work.*—The general scheme of the tests was to generate steam in a boiler having a single water tube so arranged that water circulation occurred under conditions similar to those of an actual power boiler. The steam generated was condensed and weighed, and served as the basis for computing the heat absorbed by the tube. The heat was obtained by burning illuminating gas and passing the products of combustion parallel to the axis of the tube.

Sufficient data were obtained to correlate the heat transmission coefficients with the rate of gas flow, the gas temperature, the velocity of the water in the tube, the temperature of the water, and the temperature of the tube.

3. *Acknowledgments.*—Acknowledgment is made to Professors A. C. Willard, A. P. Kratz, and J. A. Polson of the Department of Mechanical Engineering, who have offered helpful suggestions in the design of the apparatus, the testing procedure, and the calculations and presentation of the results.

Mr. C. H. Cather assisted in the calibration of the Pitot tubes used for the determination of water velocity and correlated the results obtained. Mr. E. B. Boynton assisted in the determination of the insulation losses of the apparatus. Mr. L. J. Bowditch assisted in performing the tests, averaging the data, and calculating some of the preliminary results.

The work of Messrs. Cather, Boynton, and Bowditch was done while they were enrolled as members of the Graduate School, and constituted the laboratory requirement toward the degree of Master of Science in Mechanical Engineering.

The investigation has been carried on as a part of the work of the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the director, and of the Department of Mechanical Engineering, of which Prof. A. C. Willard is the head.

II. Description of Apparatus

4. *Boiler.*—The general arrangement of the testing plant is illustrated in Figs. 1 and 2.

The boiler drum was of 12-in. standard pipe slightly over 17 ft. long. The headers were of 4-in. standard pipe and the bends were of 4-in. extra heavy pipe. The 4-in. No. 10 B.W.G. boiler tube constituting the heating surface was 10 ft. ¾ in. long, giving a gas side heating surface of 10.55 sq. ft. After forged steel flanges were welded to the tube, it was bolted in place to the header flanges. The slope of the tube was 21 degrees to the horizontal.

Steam leaving the top of the drum entered a steam separator draining to the bottom of the drum. From the separator the steam passed through the pressure control valves into the condenser.

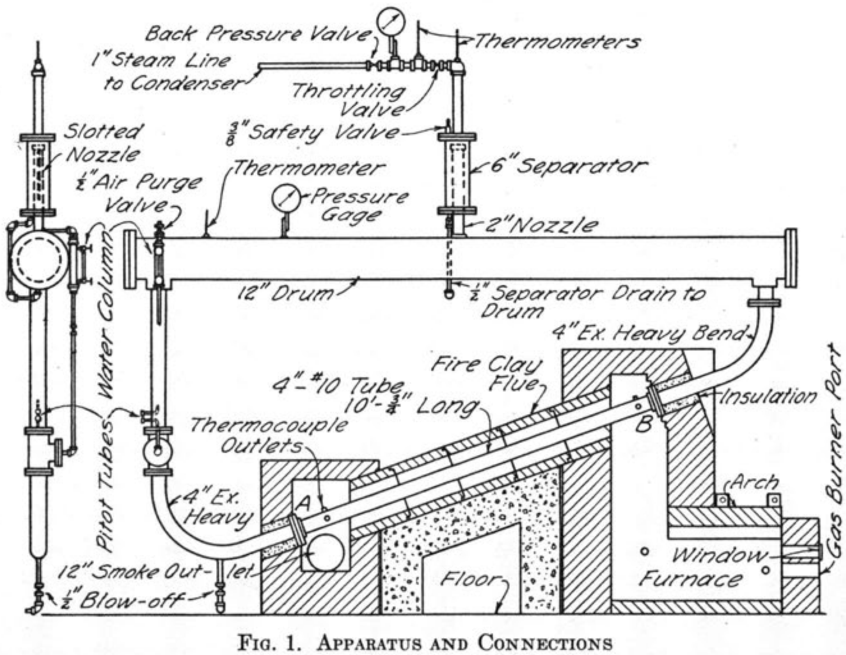

Fig. 1. Apparatus and Connections

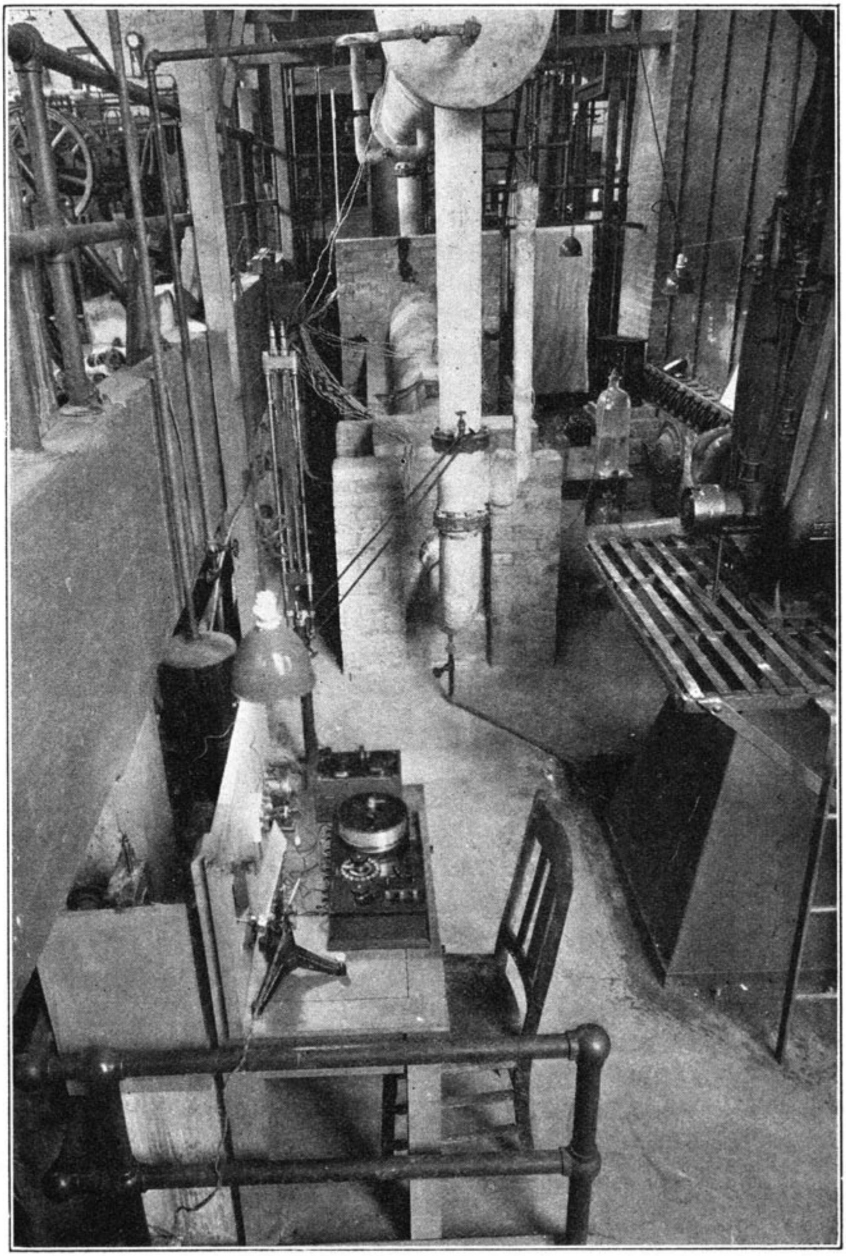

FIG. 2. REAR END VIEW OF APPARATUS

All parts of the boiler, with the exception of the test tube and flanges, part of the water column, and those flange joints which might have to be broken, were insulated with 85 per cent magnesia. The steam separator and lead were likewise insulated for 18 in. beyond the back pressure valve. The remainder of the condenser steam lead was left bare intentionally to assist the condenser in its function.

5. *Condenser and Feed Pump.*—The condenser was a double-pipe, counter-flow condenser, with steam entering the upper end. The feed pump was used to recharge the boiler with water only at the end of tests when the water level was low.

6. *Furnace and Flue.*—The gas-fired furnace, with arched top and stays, was constructed of fire-brick. It was 18 in. wide, 20 in. high, and 36 in. long. The 18-in. square uptake and the smoke connection chamber were also made of fire-brick. The fire-clay gas flue surrounding the water tube had an internal diameter of 12 in., giving an effective gas passage area of 0.716 sq. ft. after subtracting the cross-sectional area of the water tube.

The flue gases were discharged from the setting by a 12-in. diameter smoke connection shown in Fig. 2. This connection was fitted with a damper and discharged the gases at the bottom of a 40 in. by 40 ft. chimney.

7. *Gas Burner.*—The gas burner shown in Fig. 3 was made of a 4-in. tee and burner pipe. The 1-in. primary air line entered the $1\frac{1}{2}$-in. gas line in the burner, giving an inspirator effect. The secondary air entered the top outlet of the tee through a volume-measuring tube. The burner was sealed into the furnace with asphaltum cement to prevent air leakage.

8. *Orifices.*—The primary air orifice, 0.75 in. in diameter, was installed in a 1-in. pipe and was calibrated by timing the interval necessary for a known weight of air at a given pressure to pass through the orifice.*

Two gas orifices were used, as follows: No. 1, 0.5 in. in diameter was used for tests 1 to 29 inclusive; and No. 2, 0.836 in. in diameter, replaced No. 1 in tests 30 to 64, inclusive, to supply larger volumes of gas than were obtainable with orifice No. 1.

These orifices (see Fig. 3) were calibrated with air in the same manner as the primary air orifice; when used with gas, corrections were applied for the difference in density of the two fluids.

*The air weighing plant used is described completely in "Investigation of Warm-Air Furnaces and Heating Systems." Univ. of Ill. Eng. Exp. Sta. Bul. 120.

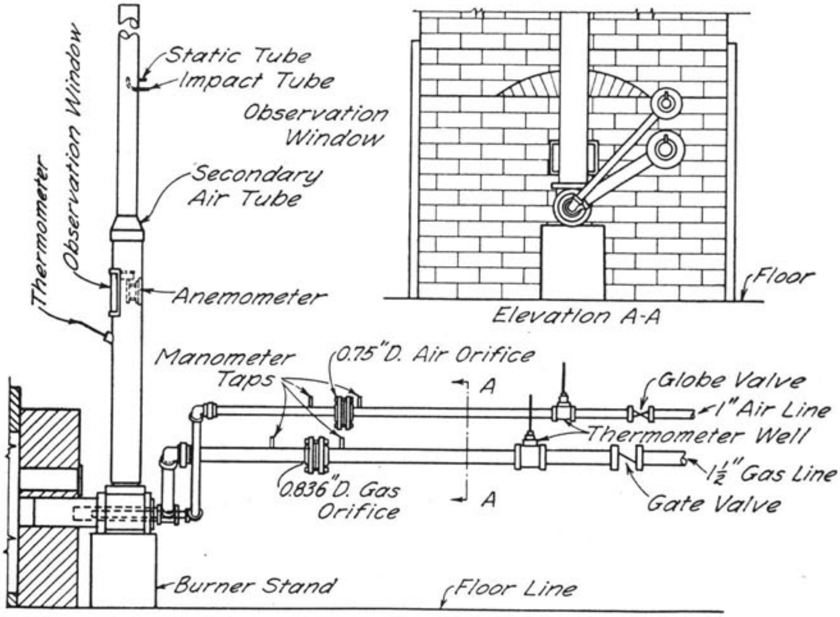

FIG. 3. DETAIL OF BURNER, ORIFICES, AND SECONDARY AIR TUBE

9. *Secondary Air Tube.*—The secondary air tube (see Fig. 3) consisted of two parts: the lower or anemometer section, which was 3 ft. 1 in. long and 4.44 in. in diameter; and the upper or Pitot tube section, which was 24 in. long and 3¼ in. in diameter.

The anemometer section consisted of a tube in which an anemometer was rigidly fastened with the rotor and guard facing upstream, or toward the top. The tube was fitted with a mica observation window, through which readings could be taken, and an external extension to the anemometer clutch for stopping the recording mechanism.

The Pitot tube section consisted of a straight pipe with a static tube and a rigidly centered impact, or Pitot, tube facing upstream. The Pitot tube was located 2 ft. from the top of the pipe.

Both the secondary air measuring devices were calibrated in place in the burner apparatus which was afterwards installed in the furnace. The calibration was done under an induced air flow with conditions practically the same as when the burner was in use, with the exception that there was no illuminating gas flow and no flame; consequently the flame effect on the air induction was not introduced.

10. *Pitot Tubes for Water Velocity.*—The water velocity in the "downcomer" header of the apparatus was determined by two Pitot tubes connected to a Venturi meter manometer. One tube pointed upstream and the other downstream.

These tubes were calibrated in place, using a kerosene-diluted solution of chlorabenzene (C_6H_5Cl) as a manometer fluid, and water at three different temperatures: namely, 56, 121, and 166 deg. F.

The expression

$$V = \frac{1.032\, h^{0.463}}{t^{0.0935}} \tag{1}$$

where

V = water velocity in ft. per sec.
h = reading of manometer in in. of chlorabenzene at 70 deg. F.
t = water temperature in deg. F.
was found to express the results obtained and was used to compute the water velocities on all tests.

11. *Thermometers.*—The room, Pitot tube manometer, gage board, primary and secondary air, gas, drum steam, and throttling valve temperatures were taken with mercury thermometers calibrated by comparison with a certified Bureau of Standards thermometer. The steam temperature thermometers were calibrated in contact with a Bureau of Standards thermometer in oil, and stem corrections were applied in all cases.

12. *Thermocouple System.*—The locations of the thermocouples used are shown in Fig. 4. Two twenty-point switches were used, one of which is called "East," and the other "West," and the couples will hereafter be referred to as "E" or "W" according to the respective switch connection.

A single cold junction circuit of thermocouples was used.* Since two switches were employed, two cold junctions were necessary and were placed in Dewar flasks containing melting ice.

All couples were insulated and run to a common junction box where they were connected by binding posts to the individual leads of 25-pair telephone cables running to the two selective switches. Any stray e.m.f. caused by a temperature change in the junction box connections was compensated for by an opposed e.m.f. due to the compensating-couple circuit running from the switches through the junction box to the cold junction.

*Foote, Fairchild, and Harrison, "Pyrometric Practice," Bureau of Standards, Paper No. 170.

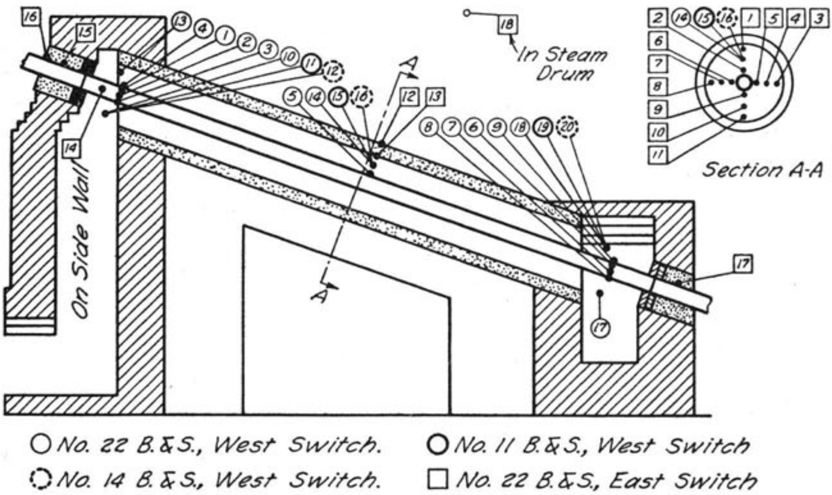

FIG. 4. THERMOCOUPLE LOCATIONS

Chromel-alumel thermocouple wire of one of three sizes, namely, No. 22, 14, or 11 B&S wire, was used, depending upon the purpose. All couple ends were twisted and acetylene-welded to a ball tip with borax as a flux.

Sample couples of each size of wire were sent to the Bureau of Standards for calibration above 500 deg. F. Those couples used for measuring water and steam temperatures were individually calibrated in boiling oil in contact with a Bureau of Standards thermometer.

The couples used to determine the temperature of the water in the tube were shaped as shown in Fig. 5, the top right-hand section showing the tube outlets in the position in which they were used. These couples were formed to give a 2-inch extension up stream in order to neutralize the heat conduction from the flue gas and tube wall to the couple junction. Twelve coats of insulating enamel were baked on after the couple leads had been insulated from each other by small glass beads. The insulated couples were then tested with a sensitive galvanometer and dry cells for electrical leakage in boiling water, but no leakage was evident.

The couple leads were insulated from the bottom of the packing space to an inch beyond the outlet with small glass beads and from the top of the outlet to the junction box with braided asbestos insulation. Asbestos insulation was used in the packing gland. The insulating varnish probably burned off at some point in the brass gland but this left the beads as additional insulation.

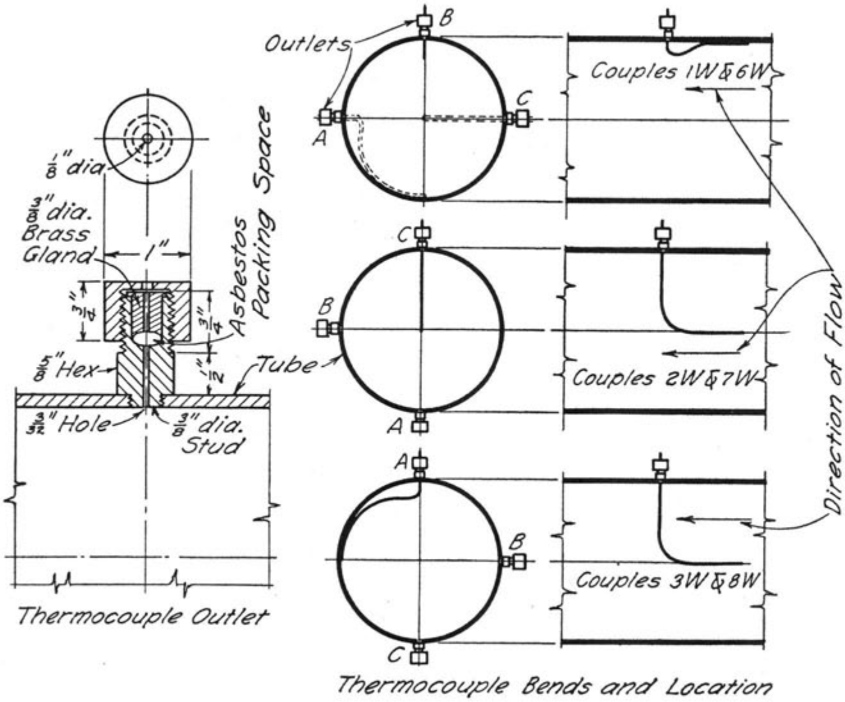

FIG. 5. DETAIL OF THERMOCOUPLES

The couples 4W, 5W, and 9W measuring the temperatures of the outside of the tube were shaped like 2W and 7W and were then insulated with enamel. A burred notch was cut in the outside surface of the tube, in which the couple was placed. The burred edge was then turned over the tube but on account of the shortness of the burr, it did not completely cover the couple. Number 32 B&S iron wire was wrapped tightly around the tube. This held the couple securely in place, and at the same time served as a reinforcement for the cement with which the couple was covered and protected from the direct action of the flue gas. The couples were protected by glass beads and small mounds of cement at the points where they left the tube. Braided asbestos insulation was used to the junction box.

The tube was given a cold hydrostatic test in place and at the same time the couples were given an insulation test with a galvanometer and dry cells. No water or electrical leakage was evident.

The wire size and location of the various couples were:

Couple No.	Wire Size (B&S)	Location
4W, 5W, 9W	22	Tube surface, gas side
1W, 2W, 3W, 6W, 7W, 8W	22	Water temperature in tube
10W, 13W, 14W, 17W, 18W	22	Flue gas temperature
11W, 15W, 19W	11	Flue gas temperature
12W, 16W, 20W	14	Flue gas temperature
1E to 11E, incl.	22	Flue gas temperature
12E	22	Flue surface, air side
13E	22	Flue surface, gas side
14E	22	Brickwork temperature
15E, 16E, 17E	22	Outside pipe surface
18E	22	Thermometer well in steam drum

13. *Potentiometers.*—Potentiometers were used to determine the e.m.f. of all couples. For the west switch couples (see foreground of Fig. 2) a Leeds and Northrup precision potentiometer was used in conjunction with a Leeds and Northrup laboratory galvanometer. This instrument reads directly to 0.005 millivolts as used, or an equivalent of about 0.2 deg. F. with chromel-alumel wire.

A Leeds and Northrup portable potentiometer was used at the east switch, and was checked within its reading limit from time to time with the precision instrument. The portable instrument read 0.1 millivolt directly up to 14 m.v. (630 deg. F.) and 0.5 millivolt above this point to 70 m.v.

III. Testing Procedure

14. *Testing Procedure.*—The data presented are the results of 64 tests made in 19 groups. Each series is lettered and each test is numbered consecutively as noted in Table 1.

Each series usually consisted of four separate tests. For each series the gas flow and temperature were maintained constant, and for each successive test of any series the steam pressure was increased between tests. The steam pressure steps were usually 10, 50, 100, and 140 lb. per sq. in. gage.

The test procedure was as follows: The apparatus was filled with water and the water column level and water temperature noted. The total weight of water in the apparatus was then known from the water volume calibration of the apparatus before the test was begun.

The gas burner was lit and the gas and air flow adjusted to give the gas temperature and rate of flow desired. The apparatus and water were heated with the discharge valve closed until the desired test steam pressure was obtained. This warming-up period required from an hour to three hours depending on the rate of gas flow and the gas temperature. When a positive drum pressure was reached, the drum was purged of all air.

When the first test steam pressure (usually 5 lb. gage) was reached, the back pressure valve was opened wide and the throttling valve was opened and adjusted so that the steam pressure remained constant; the amount of steam discharged during this adjusting period was condensed and weighed. When constant steam pressure was obtained and the apparatus had attained temperature equilibrium, the test was started and the steam generated during the test period was condensed and weighed. This test period was usually an hour. When the test period was completed the throttling valve was closed and the steam pressure allowed to increase to the next test pressure, usually 50 lb. gage. The throttling valve was then adjusted to give constant steam pressure and the steam generated under constant conditions for the test period was again condensed and weighed. The remaining tests (usually at 100 and 140 lb. gage) were conducted in a similar manner. The condensed steam discharged between tests during the pressure adjusting period was always weighed and recorded in order that the weight of water in the apparatus at the beginning and the end of the test could be calculated. This was necessary to determine the amount of heat added to the water in the apparatus if the steaming pressure was increased, since this heat did not appear in the steam condensed. In other words, this was necessary to determine the net change in the heat of the liquid of the water in the apparatus if the pressures (and saturation temperatures) at the beginning and end of the test were not the same. It was rather difficult to keep the pressure constant within a variation of less than 2 lb., and therefore a pressure increase was permitted rather than a pressure decrease, as "flashing" of water into steam in the drum naturally occurred with a decrease in pressure.

The maximum length of the test period was controlled by the weight of condensed steam necessary to reduce the weighing error. The scales used were correct to and could be read to 0.25 lb. and, therefore, to limit the weighing error to one per cent, all tests were run until at least 25 lb. of condensed steam were collected. The test periods varied from forty-five minutes to two hours with the majority of the tests lasting one hour. Five of the tests continued for less than an

TABLE 1

CALCULATIONS OF HEAT ADDED TO TEST TUBE

Series	Test No.	Duration hr.	Heat to Steam—B.t.u.							Heat to Water—B.t.u.				Heat Added to Metal During Test H_m	Insulation Loss B.t.u. per hr. H_i	End Correction B.t.u. per hr. H_e	Total Heat Added to Tube B.t.u. per hr. H_t	Per Cent "Builder's Rating"
			Lb. Condensed Steam-Test	Abs. Steam Pressure lb. per sq. in.	Drum Temperature deg. F.	Quality Steam	Total Heat Above 32 deg. F.	Net Heat Added	Total Heat Added to Steam During Test H_s	Wt. of Water in Apparatus at End of Test lb.	Initial Temperature deg. F.	Final Temperature deg. F.	Total Heat Added to Water During Test H_w					
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
A	2	1.0	52.25	28.5	247.0	0.992	1156.1	940.6	49 190	850	247.3	250.2	2 465	592	10 620	100	62 970	178
	3	0.767	21.0	64.9	298.3	0.978	1160.6	901.6	18 930	800	290.0	304.9	11 920	3 040	16 520	100	60 820	172
B	4	1.217	68.25	64.6	298.1	0.990	1170.8	903.2	61 610	768.7	298.4	301.2	2 152	572	17 950	−100	60 660	171
	5	0.85	49.00	144.4	355.7	0.990	1185.1	859.6	42 100	718.0	354.3	356.9	1 865	530	22 890	−200	75 040	212
C	6	1.833	64.50	23.3	238.0	0.990	1151.2	945.3	60 990	834.5	237.7	241.9	3 507	857	9 460	300	45 410	128
	7	1.25	59.60	63.2	296.7		1170.9	902.5	53 800	766.4	299.2	298.1	−844	−224	16 180	200	58 560	165
	8	1.10	45.25	116.3	339.4		1181.9	872.4	39 490	713.7	339.0	339.9	642	184	19 900	100	56 670	160
	9	1.083	45.50	155.4	361.0		1186.6	853.6	38 830	667.0	361.5	361.2	−200	−61	21 870	100	57 570	163
D	10	1.467	55.0	31.5	254.3	0.989	1156.8	930.6	51 200	822.7	257.8	257.8	0	0	12 450	100	47 440	134
	11	1.467	57.5	57.8	290.5	0.989	1168.8	902.3	51 900	745.7	297.4	294.8	−1 937	−531	16 490	−100	50 090	141
	12	1.417	42.25	120.0	341.8	0.989	1181.5	868.2	36 680	698.0	342.6	343.8	838	245	21 150	100	47 870	135
	13	0.675	15.25	159.0	362.5	0.984	1182.0	848.3	12 940	679.0	362.2	366.2	2 716	816	22 840	−100	47 140	133
E	14	1.00	36.25	21.7	246.5	0.989	1148.8	929.6	33 700	750.2	250.9	250.2	−525	−143	10 300	500	43 830	124
	15	1.00	22.50	62.9	296.9		1169.9	904.5	20 350	723.0	296.3	302.3	4 338	1 224	16 200	131	42 240	119
	16	1.00	21.80	114.5	338.3		1181.0	870.3	18 980	694.0	340.2	341.2	694	204	20 080	−62	39 900	113
	17	0.404	4.25	158.11	361.5		1186.0	850.1	3 611	681.0	364.3	366.0	1 158	347	22 200	−293	34 510	98
F	18	1.00	49.75	20.5	229.2	0.996	1155.6	952.8	47 400	889.2	234.7	232.2	−2 222	−510	8 100	66	52 830	149
	19	1.00	23.75	74.9	308.5	0.997	1181.8	908.5	21 590	840.4	304.0	317.3	11 180	2 713	17 220	−28	52 680	149
	20	1.00	31.4	118.7	340.8		1187.4	874.9	27 480	784.0	341.9	345.8	3 056	796	20 280	−387	51 230	145
	21	1.00	33.0	159.2	363.0		1192.6	856.1	28 260	726.5	364.8	366.8	1 453	408	22 120	−267	51 970	147

TABLE 1 (CONTINUED)

CALCULATIONS OF HEAT ADDED TO TEST TUBE

Series	Test No.	Duration hr.	Heat to Steam—B.t.u.							Heat to Water—B.t.u.				Heat Added to Metal During Test H_m	Insulation Loss B.t.u. per hr. H_i	End Correction B.t.u. per hr. H_e	Total Heat Added to Tube B.t.u. per hr. H_t	Per Cent "Builder's Rating"
			Lb. Condensed Steam-Test	Abs. Steam Pressure lb. per sq. in.	Drum Temperature deg. F.	Quality Steam	Total Heat Above 32 deg. F.	Net Heat Added	Total Heat Added to Steam During Test H_s	Wt. of Water in Apparatus at End of Test lb.	Initial Temperature deg. F.	Final Temperature deg. F.	Total Heat Added to Water During Test H_w					
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
G	22	1.00	58.5	20.6	231.3	0.997	1155.4	955.0	55 850	827.0	232.3	233.1	662	163	7 876	−19	64 530	183
	23	1.00	56.0	64.3	297.8	0.997	1177.7	908.6	50 900	711.5	299.9	298.4	−1 067	−306	15 820	−227	65 120	184
	24	1.00	49.8	124.1	344.2	0.997	1189.0	873.9	43 500	601.7	344.4	345.7	782	265	20 090	−343	64 290	182
	25	0.75	36.75	156.7	361.6	0.997	1192.5	857.3	31 510	504.9	363.6	361.1	−1 262	−510	21 560	−337	60 900	172
H	26	1.5	19.13	23.8	241.0	0.987	1151.5	948.3	18 140	925.0	235.1	244.3	8 510	1 877	10 900	155	30 080	85
	27	1.5	18.63	65.7	299.8	0.998	1179.6	911.8	16 980	909.5	298.6	299.8	1 091	245	17 560	134	29 900	85
I	28	1.5	26.88	22.9	239.4	0.989	1152.0	949.8	15 510	722.0	234.1	242.0	5 700	1 611	9 928	−35	25 110	71
	29	1.5	20.63	81.1	314.2	0.998	1185.7	903.3	18 640	693.5	312.8	316.2	2 357	694	18 120	95	32 680	92
J	30	1.0	84.0	55.8	287.0	0.998	1175.2	928.5	78 000	820.0	278.0	292.6	11 970	2 980	14 720	154	107 820	305
	31	1.0	99.0	116.0	339.0	0.998	1189.3	879.2	87 100	662.0	339.6	340.4	530	163	19 320	−1348	105 670	299
	32	0.75	71.5	159.5	363.5	0.998	1194.4	860.2	61 500	547.0	362.6	366.8	2 297	857	21 300	−1224	106 300	300
K	33	1.0	72.0	31.6	255.0	0.998	1165.1	944.2	68 000	835.0	252.6	259.9	6 096	1 490	11 560	84	87 230	247
	34	1.0	73.5	70.4	303.8	0.998	1180.0	908.3	66 780	742.5	302.4	306.6	3 119	857	16 720	−604	86 870	246
	35	1.0	66.75	129.7	346.0	0.998	1190.1	877.2	58 570	670.7	342.3	347.7	3 620	1 102	20 200	−350	83 140	235
	36	1.0	73.63	158.3	362.9	0.997	1193.3	859.5	63 270	570.0	362.3	362.3	0	0	21 300	−480	84 090	238
L	37	1.0	154.5	21.3	233.4	0.998	1158.2	954.0	147 400	755.0	236.1	237.8	1 283	347	8 620	−726	156 920	444
	38	1.0	164.5	73.1	309.0	0.998	1182.1	904.2	148 800	800.0	308.4	313.8	4 320	1 102	17 380	1689	173 290	490
	39	1.0	160.5	115.3	339.5	0.998	1189.6	881.4	141 500	765.0	337.8	341.9	3 136	837	19 780	−127	165 130	467
	40	1.0	169.0	153.8	361.3	0.999	1195.4	863.1	145 900	650.0	360.8	363.5	1 755	551	21 420	−668	168 960	477
M	41	1.0	117.8	27.3	248.4	0.992	1157.8	938.6	110 500	602.2	250.9	257.3	3 855	1 306	11 120	530	127 310	360
	42	1.0	158.0	70.8	304.3	0.998	1180.9	901.3	142 400	432.2	310.1	316.4	2 722	1 285	17 000	1352	164 760	461
	43	1.0	139.0	114.2	336.7	0.998	1189.0	878.5	122 100	660.5	340.0	347.8	5 150	1 592	19 600	811	149 250	422
	44	1.0	157.5	162.1	362.3	0.999	1195.3	853.6	134 400	489.0	369.7	371.9	1 076	449	21 800	698	158 450	448

TABLE 1 (CONCLUDED)

CALCULATIONS OF HEAT ADDED TO TEST TUBE

Series	Test No.	Duration hr.	Heat to Steam—B.t.u.							Heat to Water—B.t.u.				Heat Added to Metal During Test	Insulation Loss B.t.u. per hr.	End Correction B.t.u. per hr.	Total Heat Added to Tube B.t.u. per hr.	Per Cent "Builder's Rating"
			Lb. Condensed Steam-Test	Abs. Steam Pressure lb. per sq. in.	Drum Temperature deg. F.	Quality Steam	Total Heat Above 32 deg. F.	Net Heat Added	Total Heat Added to Steam During Test H_s	Wt. of Water in Apparatus at End of Test lb.	Initial Temperature deg. F.	Final Temperature deg. F.	Total Heat Added to Water During Test H_w	H_m	H_i	H_e	H_t	
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
N	45	1.0	141.5	21.8	234	0.998	1157.4	945.1	133 700	495.5	240.1	241.4	644	265	7 900	−500	142 000	401
	46	1.0	118.8	66.4	300.1	0.999	1179.5	909.7	108 000	569.2	300.6	304.2	2 048	735	15 050	−1432	124 400	352
	47	1.03	144.5	119.4'	341.7	0.999	1190.4	879.4	127 100	573.5	340.5	346.1	3 210	1 143	18 950	−985	145 270	411
	48	0.76	107.8	158.0	363.3	0.999	1195.3	862.7	93 000	607.0	361.1	368.6	4 555	1 530	20 720	−2329	147 490	417
O	49	1.0	47.4	26.5	245	0.987	1160.0	946.3	44 880	895.1	245.5	259.9	6 621	1 510	11 960	−164	64 810	183
	50	1.0	44.75	69.6	303.5	0.996	1178.7	905.2	40 500	843.2	304.2	308.5	3 624	878	17 710	−267	62 440	177
	51	1.0	44.3	118.0	341.5	0.996	1187.5	874.7	38 740	788.2	342.2	346.1	3 074	796	21 150	−540	63 220	179
	52	1.0	43.5	145.0	356.8	0.996	1191.1	859.4	37 390	736.5	360.3	359.9	−295	−82	22 320	−462	58 870	167
P	53	1.0	99.0	31.5	254.6	0.998	1165.4	936.7	92 750	804.5	260.3	263.5	2 572	653	10 960	−138	106 800	302
	54	1.0	111.5	69.4	303.5	0.998	1180.1	912.3	101 800	689.5	298.6	314.9	11 240	3 326	16 390	−96	132 660	375
	55	1.0	129.3	117.3	341.2	0.997	1190.2	876.3	113 300	739.2	343.2	348.2	3 696	1 020	19 510	−692	136 830	387
	56	1.0	134.8	145.1	357.2	0.997	1193.7	858.7	115 700	570.4	363.4	363.6	114	41	20 820	−722	135 950	384
Q	57	1.0	181.0	26.7	247.1	0.996	1161.2	941.5	170 400	637.0	251.4	256.2	3 058	980	9 860	−865	183 440	518
	58	1.0	180.0	122.5	344.2	0.998	1190.6	872.9	157 100	381.0	346.9	352.1	1 980	1 061	19 960	−889	179 210	507
R	59	1.0	141.0	29.7	251.1	0.992	1159.2	936.2	132 000	730.0	254.6	260.2	4 089	1 143	11 600	−697	148 140	419
	60	1.0	167.3	125.3	345.7	0.999	1191.0	872.1	145 900	535.7	348.0	355.1	3 800	1 449	21 040	−444	171 750	485
S	61	1.0	65.25	18.5	226.3	0.996	1154.1	957.3	62 450	734.7	228.7	229.8	808	224	7 792	−621	70 650	200
	62	1.0	60.0	65.1	299.4	0.998	1179.2	909.3	54 560	810.7	300.6	302.8	1 784	448	16 100	−1064	71 830	203
	63	1.0	53.0	106.3	333.9	0.998	1187.9	884.5	46 890	751.7	333.1	336.6	2 629	714	19 100	−1114	68 220	193
	64	1.0	49.5	146.3	357.0	0.998	1192.9	864.5	42 800	700.2	357.1	360.3	2 240	653	21 210	−1213	65 690	186

NOTES: Goodenough's Steam Table Values were used.
' mark indicates assumed quantities.
Col. 12 and 13 are averages of couples 2W, 7W, and drum temperature.

Col. 19 = Col. 18. $\div \frac{10.55}{10} \times 33.523$

hour because of some unusual event, such as blowing of safety-valve, low water level in drum, or low gas pressure.

15. *Quality of Steam Generated.*—The quality of the steam leaving the apparatus was determined by the throttling process. The drum pressure was maintained constant by means of the 1-in. throttling valve through which the steam passed to the low-pressure side. The temperature of the steam on the high-pressure side was obtained by means of a thermometer in an 8-in. oil-well. The temperature of the steam on the low-pressure side was determined by a thermometer, the bulb of which was exposed to the steam. The pressure on the high-pressure side of the throttling valve was taken as the saturation pressure of the temperature indicated by the high-pressure thermometer. The steam pressure on the low-pressure side was determined either by a mercury manometer, or by a calibrated pressure gage, depending upon the steaming rate of the apparatus. The discharge line was insulated for two feet beyond the back-pressure valve which was installed to prevent steam leakage from the apparatus during the interval between tests when the pressure was being increased.

16. *Gas Analysis.*—Usually four tests were run during a day at a constant gas-air ratio. One gas sample for the four tests was collected in a 75-liter jar and was analysed by an Orsat apparatus. If the gas-air ratio changed due to low gas pressure a separate gas analysis was taken. The sample was taken from the setting at the intake to the smoke connection, and thus included any air infiltration through the setting, which had been made as air-tight as possible with furnace cement.

The sample was taken by a pipe cross made of ½-in. pipe with $\frac{5}{64}$-in. holes, 1 in. on centers. The pipe cross measured 13 in. from tip to tip and was placed so that the sampling holes faced up stream.

For tests 48 to 64, a Uehling CO_2 recorder was used to check the Orsat analysis. The two methods of CO_2 determination agreed within 0.25 per cent of CO_2.

Flue gas samples, humidity, and barometric readings were taken between tests during the pressure adjusting period. A sample of the illuminating gas was also taken in a glass balloon to determine the density of the gas for computation of the weight of gas flowing through the burner orifice.

17. *Insulation Loss.*—To correct for the heat loss by radiation from the insulated apparatus during a test, a special series of six insulation loss tests was run before the setting was installed.

The pipe bend flanges were blanked at points A and B as shown in Fig. 1. Half-inch pipes were connected at the blow-off connection on the rear bend and at the blank flange on the front bend. The condensate was carried in these pipes to a receiver, the water level in which was maintained constant during a test.

The procedure was to introduce steam at various pressures into the apparatus and to weigh the condensate formed. The room temperature was taken at the two extremities of the apparatus and the steam pressure and temperature were recorded. No test was started until the rate of condensation had become constant. The test period varied from one hour to one hour and a half, with readings taken every five minutes. Tests were run at drum pressures of 144, 121, 65, 61, 23, and 22 lb. per sq. in. gage.

A throttling calorimeter was used to determine the quality of the entering steam. For the lower pressures, steam slightly superheated (3 to 10 deg.) was introduced into the drum because of the inability of the throttling calorimeter to indicate the quality at these pressures.

The insulation loss is shown in Fig. 6 plotted against the arithmetical mean temperature difference between the steam temperature in the drum and the average room temperature.

The insulation losses for all tests were taken from this graph. The room temperatures for the tests were taken at the same points as were used to determine the air temperatures for the insulation-loss mean temperature difference.

The insulation loss was determined with the apparatus filled with steam, while the test connections were applied with the lower portion of the drum and apparatus filled with water. However, the only difference between the two cases is the change in the heat transfer film coefficient on the inside surface of the apparatus.

Since this inside film resistance is approximately only 1/25 of the total resistance from the steam in the pipe to the pipe wall, through the pipe wall and one inch of insulation to the external air, a rather marked change in the inside film coefficient caused by a change of the fluid in the pipe from steam to water would not affect the insulation loss to any great extent.

IV. Calculations

18. *Flue-gas Temperatures.*—The flue-gas temperatures were taken with No. 22 B&S wire thermocouples. This small wire was used in order to decrease the error due to radiation to and from the surrounding setting.

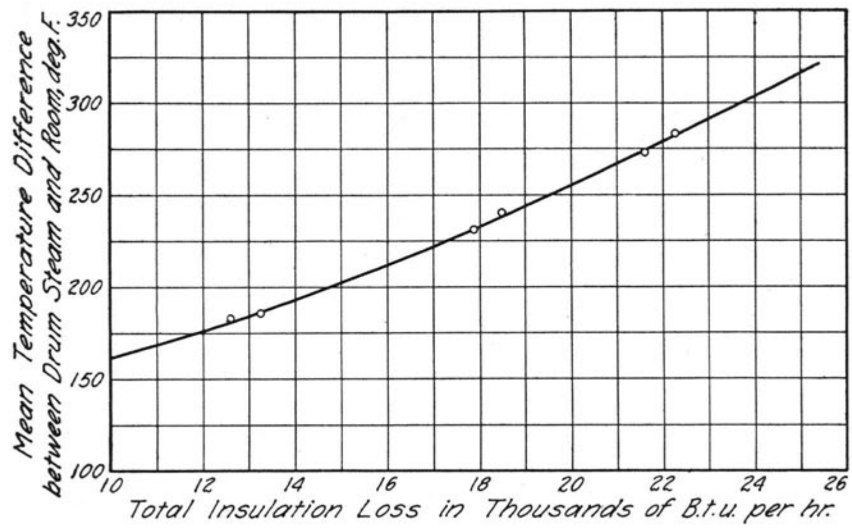

FIG. 6. INSULATION LOSS OF APPARATUS

It is evident that a thermocouple placed in a gaseous space surrounded by a solid enclosure at a lower temperature does not indicate the true temperature of the enclosed gas because of the fact that the couple junction loses heat by radiation to the cooler surrounding body; the amount of heat lost by radiation from the junction varies directly as its radiating surface; therefore, the smaller the junction, the smaller the radiation error becomes; hence, for a junction of zero radiation surface, there would be no radiation error.

A practical thermocouple used in the measurement of the temperature of gas streams involves then a radiation error. This error can be corrected by placing three or more couples of different sizes at the same point in a gas stream, noting the difference in the indicated temperature, plotting the temperature against the radiation surface of the various couples, and then extrapolating the result to get an indication of the temperature reading of a couple of zero radiating surface.*

In this work, couples 10W, 11W, and 12W were used to determine the radiation error at the high temperature end of the tube; 14W, 15W, and 16W at the center section; and 18W, 19W, and 20W at the low temperature end.

The diameter of the junction, the radiating surface, and the wire size are given in Table 2.

*"Measuring the Temperature of Gases in Boiler Settings," Bur. of Mines Bul. 145.

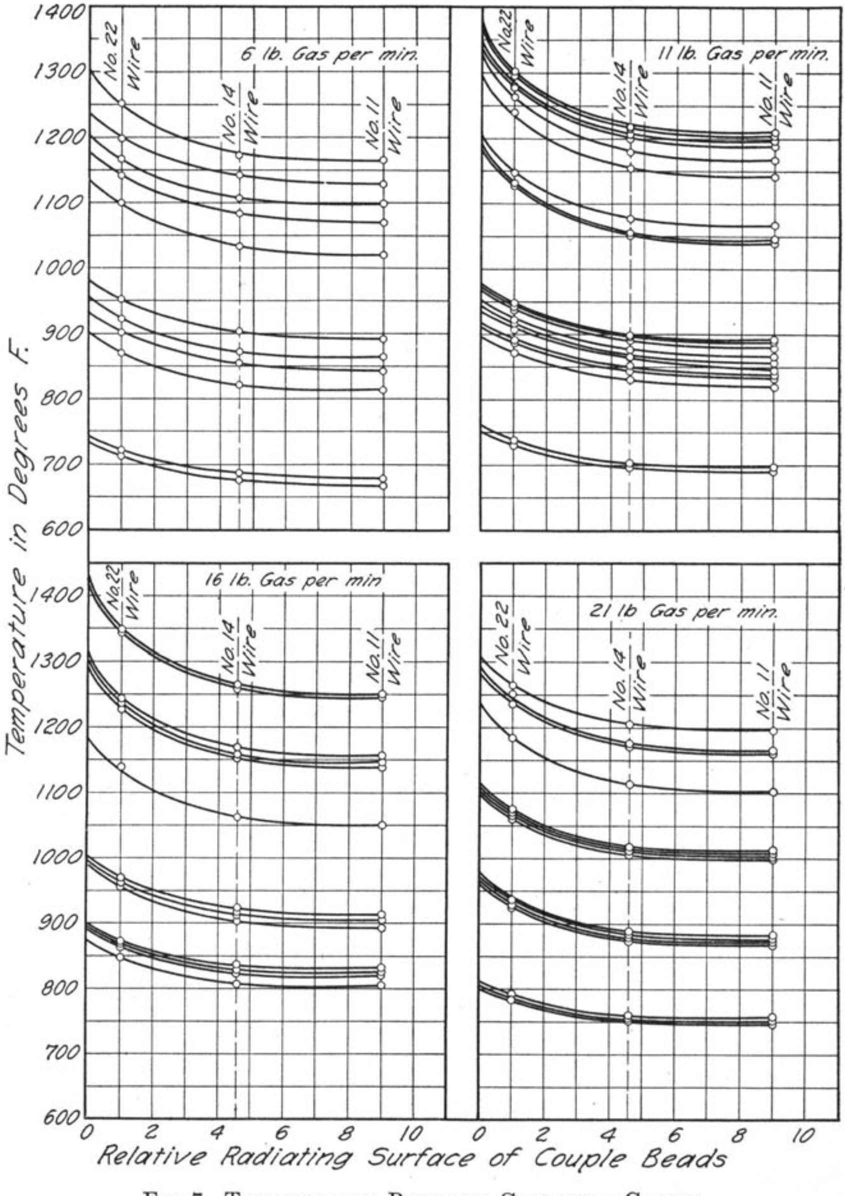

FIG. 7. THERMOCOUPLE RADIATION CORRECTION GRAPHS

TABLE 2
DIMENSIONS OF GAS TEMPERATURE COUPLES

B&S Wire Size	Diameter of Bead, in.	Radiating Surface, sq. in.	Comparative Radiating Surface No. 22 as Unity
11	0.21	0.1386	9.00
14	0.15	0.0707	4.59
22	0.07	0.0154	1.00

The radiation corrections for these tests were then divided into four groups for four different rates of gas flow (6, 11, 15, and 21 lb. per min.) and correction graphs were plotted. Corrections for any test were taken from the graph corresponding to the gas flow rate used; corrections for any couple were taken from the graph constructed from the three radiation couples nearest the couple in question.

Figure 7 illustrates the general scheme of radiation correction. These graphs were constructed from couples 18W, 19W, and 20W at the low temperature end of the setting. The abscissas are the relative radiating surfaces of the three couples and the ordinates are the indicated gas temperatures. A smooth curve was drawn through the three couple temperatures obtained for any gas temperature with the slope increasing with the gas temperature. The form of the curve from abscissa 1 to 0 is determined by conjecture but the error here is probably well within the error introduced by applying the correction of a set of couples at one point in the setting to a single couple at another point.

All flue gas temperatures were corrected by this method before calculations were made.

19. *Heat Transmitted by Tube.*—The principal problem for determination was the amount of heat energy transferred per unit of time to the test tube from the moving gas stream.

The total heat absorbed by the apparatus can be expressed by the equation

$$H_t = H_s + H_w + H_m + H_i - H_e \tag{2}$$

where

H_t = total heat energy received by water in test tube (col. 18)†

†Column numbers refer to Table 1.

H_s = heat content of generated steam which was condensed and weighed (col. 10)

H_w = heat energy added to water in boiler due to an increase in steam pressure during test (sign may be positive or negative)* (col. 14)

H_m = heat energy added to the metal of the apparatus due to an increase in the boiler temperature (sign may be positive or negative) (col 15)

H_i = heat energy loss from insulated surfaces of apparatus (col. 16)

H_e = heat energy added to water in apparatus otherwise than through test tube (sign may be positive or negative) (col. 17).

20. *Heat Content of Generated Steam, H_s.*—The heat content of the generated steam was calculated from the following expression:

$$H_s = W[xr + (i_1' - i_2')] \tag{3}$$

where

W = weight of steam in lb. condensed during test (col. 4)

x = average quality of steam generated during test (col. 7)

r = average latent heat of steam generated during test at temperature of col. 6.

i_1' = average heat of liquid during test at temperature of col. 6.

i_2' = heat of liquid of drum water at start of test at temperature of col. 12.

21. *Heat Energy Added to Water, H_w.*—The heat energy added to the water was calculated from the expression:

$$H_w = W_f \times \Delta T \times C_w \tag{4}$$

where

W_f = final weight in lb. of water in apparatus at end of test (col. 11)

ΔT = Temperature rise of water during test in deg. F. (cols. 12, 13)

C_w = specific heat of water at test temperature.

22. *Heat Energy Added to Metal, H_m.*—The heat energy added to the metal of the apparatus was calculated from the expression:

$$H_m = W_a \times \Delta T \times C_i \tag{5}$$

where

W_a = weight of apparatus in lb. (1683 lb.)

ΔT = temperature rise of apparatus during test (assumed the same as for the water)

C_i = specific heat of steel (taken as 0.12).

*To be strictly correct, the change in the heat content of the steam in the steam space during a change in pressure should be considered as well as a change in the heat content of the water in the drum; however, this change is so small that it can be safely neglected.

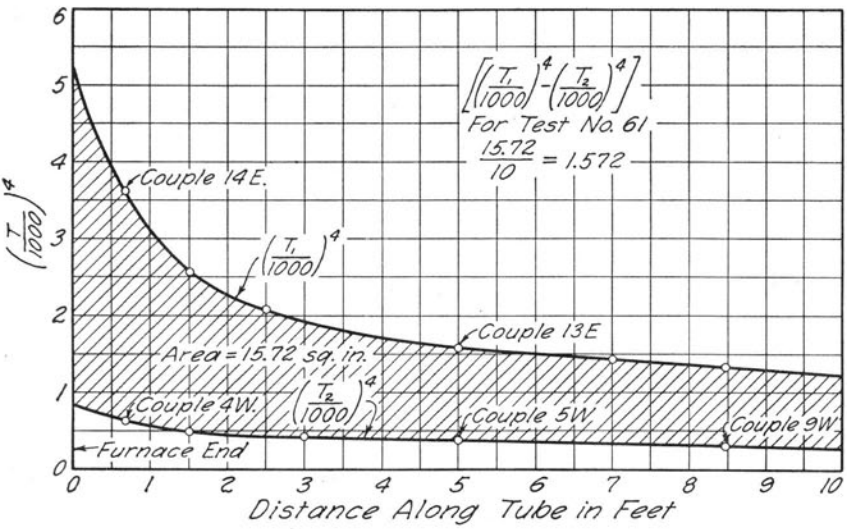

FIG. 8. METHOD OF DETERMINING RADIATION FROM FLUE

23. *Insulation Loss, H_i.*—The insulation loss was taken from Fig. 6.

24. *Heat Added Outside of Test Tube, H_e.*—Couples 15E and 17E were located on the pipe bend surfaces each at the middle of that portion passing through the setting. The temperatures determined by these couples were assumed to be the average of that portion of the pipe passing through the setting. Couple 16E was placed on the pipe surface at the point at which the pipe left the setting, to determine the temperature gradient along the pipe between the pipe wall and the water in the pipe. A comparatively accurate idea of the average temperature difference between the outside pipe surface and the water within could then be formed at those two sections where the pipe passed through the setting by means of couples 15E, 16E, and 17E.

The coefficients of heat transmission (from outside pipe surface to water) used for the end corrections were assumed to be the same as the coefficients from the test tube surface to the water for any particular test, allowance being made for the difference in metal thickness of the test tube and the header pipe. When the specific resistance (inverse of conductivity) of steel is assumed to be 581 per sq. ft. and that of the water film to be 1000, calculations demonstrate that the header pipe (0.341 in. thick) transmits 77 per cent as much heat per sq. ft. as the test tube (0.134 in. thick) with the same temperature difference between the outer surface and the water inside the pipe.

When the coefficient of heat transmission from the surface of the pipe to the water inside and the average temperature difference between the pipe surface and the water are known the total heat added at the ends can be calculated.

In practically all tests, these temperature measurements showed a heat addition at the furnace end and a heat loss at the smoke connection end. To the algebraic sum of these two quantities, a correction was made for the amount of heat supposedly radiated under item H_t and the net correction is given in col. 17, Table 1. This correction averaged about one half of one per cent of the total heat absorbed by the test tube.

Some of these corrections are positive because of the fact that the calculations showed that more heat energy was being lost at the smoke outlet end than was being gained at the furnace end. These positive corrections also indicate that more heat was being lost at the smoke outlet end installed in the setting than was lost by the same insulated radiating surface when radiating to free air without the setting. This can be explained by the fact that the asbestos and magnesia insulation used at the pipe entrance sections was rammed into place to decrease the air leakage along the header pipes into the gas passage, and therefore the conductivity of the insulation itself was undoubtedly increased.

25. *Radiant Heat Absorption.*—All of the heat energy received by the test tube was not transmitted directly by convection and radiation from the flue gases, as the fire-clay flue was at a higher temperature than the test tube, and therefore radiant energy was transferred from the inner surface of the flue to the tube.

The total heat received by the tube can be separated into two parts: (a) that received by flue-gas convection and radiation; and (b) that received by radiation from the inner surface of the flue.

The total heat energy absorbed by the tube can then be expressed as follows:

$$H_t = H_c + H_r \tag{6}$$

where

H_t = total heat energy received by tube in B.t.u. per hr.

H_c = heat energy received by convection and radiation from the moving gas stream in B.t.u. per hr.

H_r = radiant heat energy received by tube from flue in B.t.u. per hr.

H_r can be calculated fairly accurately from a form of the Stefan and Boltzmann law if the radiation coefficients and temperatures of the radiating and absorbing bodies are known. If the radiating surface completely encloses the absorbing body, and if the radiation is so-called "black-body" radiation, this law can be expressed by the equation

$$Q = S\frac{e_1e_2}{E}\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right]^* \tag{7}$$

where

Q = heat radiated in B.t.u. per hr.
S = absorbing surface in sq. ft.
e_1 = emissivity coefficient of enclosure
e_2 = emissivity coefficient of absorbing body
E = emissivity coefficient of black body
T_1 = absolute temperature deg. F. of enclosure
T_2 = absolute temperature deg. F. of absorbing body

This expression is well adapted for the calculation of H_r for this particular apparatus as the radiating surface practically encloses the absorbing body; assuming that "black-body" radiation does exist (which is probably near the truth), the only questionable factors, then, are the emissivity coefficients and the body temperatures.

The emissivity coefficients were taken from the 24th edition of Hütte's Ingenieur's Taschenbuch and were as follows:

$e_1 = 1510$ (wrought iron)
$e_2 = 1540$ (brickwork)
$E = 1650$ (black-body)

The tube absorbing surface (gas side) was 10.55 sq. ft. Equation (7) then becomes

$$H_r = 14\ 870\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right] \tag{8}$$

The method used to determine the value within the brackets in this equation for each test was as follows: The temperature of the surface of the casing was known at two points (13E and 14E, Fig. 4) and the tube temperature was known at three points (4W, 5W, and 9W). These temperatures and the flue-gas temperatures taken at 10W, 14W, and 18W were plotted on logarithmic paper with temper-

*Barker and Kinoshita, "The Effects of the Shape and Surroundings of a Hot Surface on the Radiation From It;" Univ. of London Bul. No. 1.

atures as ordinates and distances along the tube from the furnace to the couple in question as abscissas.

Since both the flue-gas and tube temperatures plotted on this basis gave straight line relations, the assumption was made that the inside surface temperature of the flue would likewise follow a straight line relation.*

Temperatures taken from the straight line logarithmic relations were converted to absolute temperatures divided by 1000, the resulting value raised to the fourth power and plotted as shown in Fig. 8, with $\left(\frac{T}{1000}\right)^4$ as ordinate and distance along the tube as abscissa. The cross-hatched area was determined by a planimeter and divided by the length which resulted in the average value of $\left[\left(\frac{T_1}{1000}\right)^4-\left(\frac{T_2}{1000}\right)^4\right]$ to be used in equation (8).

The calculated values of these various quantities for each test are given in Table 3. Column (1) indicates the average values of $\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4$. Column (2) was calculated from Column (1) and equation (8). Column (3) was calculated from equation (6) by subtracting H_r (col. 2, Table 3) from H_t (col. 18, Table 1).

This average fourth power temperature difference was affected by two variables: namely, the steam pressure for any test, and the temperature of the setting. The burner was often shut off at mid-day with a subsequent cooling of the setting. This fact accounts for the decrease in col. 1 usually between the second and third tests of any series.

26. *Heat Transmission Coefficients.*—Two sets of heat transmission coefficients have been calculated: namely, a "total" coefficient calculated from H_t and indicated as K_t in Table 3; and a "convection" coefficient calculated from H_c and indicated as K_c in Table 3.

K_t was calculated from the expression

$$K_t = \frac{H_t}{S \times \Delta T} \tag{9}$$

where

K_t = total coefficient of heat transmission in B.t.u. per sq. ft. per deg. per hr.

*This assumption was necessary because of the fact that the casing temperature was taken only at two points and therefore any form of curve might satisfy those two points when plotted; while with the gas and tube temperatures three points were determined which, when plotted, defined a straight line.

H_t = total heat energy received by test tube in B.t.u. per hr. (Table 1)

S = gas side surface of test tube in sq. ft. (10.55 sq. ft.)

ΔT = logarithmic mean temperature difference between flue gas and water in the test tube in deg. F. (col. 5, Table 3)

K_c was calculated from equation (9) by substituting H_c (convection heat energy received by tube in B.t.u. per hr., Table 3) for H_t.

ΔT was calculated from the expression

$$\Delta T = \frac{\Delta t_1 - \Delta t_2}{\log_e \dfrac{\Delta t_1}{\Delta t_2}} \tag{10}$$

where

Δt_1 = arithmetical temperature difference in deg. F. between flue gas and water at furnace end of tube (10W — 2W).

Δt_2 = same as above at smoke outlet end of tube (18W — 7W).

Couples 2W and 7W were considered as giving the average water temperatures at these points because they were centrally located in the tube.*

The average gas temperature (col. 4, Table 3) was determined by adding the average water temperature to the logarithmic mean temperature difference (col. 5). This value is useful for comparing and applying results.

27. *Water Velocity.*—The water velocity (col. 19, Table 3) was calculated from the calibration equation (1), the water temperature being given by 7W, since the actual difference in temperature at the Pitot tubes and couple 7W never exceeded three degrees. This velocity was at the Pitot tube section in the down-comer. The water velocity in the tube was greater than this by the ratio of the cross-section of the header to that of the tube, or by

$$\frac{0.0884}{0.0762} = 1.16$$

28. *Specific Volume of Flue Gas and Average Gas Velocity.*—The specific volume of the flue gas was determined by assuming complete combustion of the hydrogen in the illuminating gas to determine the weight of water formed by the combustion of hydrogen; and that the flue gas contained only CO_2, CO, O_2, N_2 and water vapor.

The average illuminating gas analysis was determined from a number of tests by the Chemistry Department of the University of

*After test 31, couples 2W and 7W read from 2 to 4 deg. high, but the error introduced in the mean temperature difference is of the order of one-half of one per cent or less.

TABLE 3

CALCULATIONS AND RESULTS

Series	Test No.	Av. $\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right]$	Heat by Radiation B.t.u. per hr. H_r	Heat by Convection B.t.u. per hr. H_c	Average Flue Gas Temperature deg. F.	Log. Mean Temp. Diff. T	Total Coef. of Heat Trans. B.t.u. sq. ft. deg. hr. K_t	Convection Coef. of Heat Trans. B.t.u. sq. ft. deg. hr. K_c	Lb. Flue Gas per min.	Flue Gas Properties: Lb. Water Vapor per min.	% by Vol. CO_2	% by Vol. O_2	% by Wt. CO_2	% by Wt. O_2	% by Wt. N_2	% by Wt. H_2O	Specific Vol. cu.ft. per lb.	Total cu. ft. Gas per min.	Gas Vel. ft. per sec.	Water Velocity ft. per sec.
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
A	2	1.630	24 240	38 730	1109	859	6.950	4.274	6.46	0.489	6.4	9.9	9.0	10.0	74.0	7.6	42.15	272	6.34	1.09
	3	1.676	24 910	35 910	1152	854	6.756	3.988	6.43	0.507							43.30	278	6.48	1.44
B	4	2.154	32 060	28 600	1180	881	6.522	3.074	6.73	0.477	7.8	10.0	10.72	10.0	72.5	7.26	43.40	293	6.81	1.05
	5	2.246	33 400	41 640	1219	863	8.240	4.575	6.81	0.502							44.45	306	7.11	1.12
C	6	1.483	22 050	23 360	1043	802	5.370	2.760	9.89	0.538	4.5	14.3	6.42	14.8	73.6	5.5	40.12	397	9.23	0.58
	7	1.626	24 190	34 370	1068	770	7.205	4.230	9.71	0.539							38.12	370	8.61	1.08
	8	1.600	23 800	32 870	1084	744	7.220	4.187	9.67	0.527	3.9	15.1	5.57	15.65	73.5	5.71	41.81	404	9.41	1.41
	9	1.646	24 490	33 0[illegible]0	1097	735	7.419	4.260	9.54	0.567							42.04	401	9.33	1.33
D	10	1.093	16 250	31 190	982	724	6.210	4.080	16.52	0.554	2.8	16.9	4.09	17.95	74.8	3.34	37.90	626	14.6	0.868
	11	1.083	16 110	33 980	997	702	6.761	4.585	16.97	0.567							38.29	650	15.1	1.09
	12	1.062	15 800	32 070	995	651	6.960	4.665	16.84	0.616	2.6	16.9	3.79	17.95	74.8	3.67	38.39	647	15.1	1.34
	13	1.058	15 740	31 400	996	632	7.070	4.710	16.75	0.618							38.41	644	15.0	1.54
E	14	0.991	14 740	29 092	899	645	6.440	4.272	21.70	0.805	2.4	16.9	3.53	18.10	74.9	3.66	35.48	770	17.9	1.23
	15	0.931	13 850	28 393	900	601	6.660	4.475	21.66	0.802							35.50	770	17.9	1.39
	16	0.867	12 900	26 996	896	555	6.810	4.610	21.71	0.779							35.41	770	17.9	1.83
	17	0.828	12 310	22 197	902	537	6.090	3.914	21.47	0.777							35.57	764	17.8	1.98
F	18	1.501	22 320	30 514	1039	804	6.235	3.600	13.44	0.577	2.3	19.3	3.19	20.78	72.45	4.21	39.16	527	12.3	1.48
	19	1.426	21 200	31 475	1092	781	6.395	3.820	13.52	0.566							40.55	549	12.8	1.70
	20	1.295	19 210	32 015	1065	710	6.842	4.275	13.75	0.582							39.85	548	12.8	1.75
	21	1.212	18 030	33 944	1064	695	7.081	4.630	13.61	0.557							39.82	544	12.7	1.68

TABLE 3 (CONTINUED)

CALCULATIONS AND RESULTS

Series	Test No.	$Av.\left[\left(\frac{T_1}{1000}\right)^4-\left(\frac{T_2}{1000}\right)^4\right]$	Heat by Radiation B.t.u. per hr. H_r	Heat by Convection B.t.u. per hr. H_c	Average Flue Gas Temperature deg. F.	Log. Mean Temp. Diff. T	Total Coef. of Heat Trans. B.t.u. sq. ft. deg. hr. K_t	Convection Coef. of Heat Trans. B.t.u. sq. ft. deg. hr. K_c	Lb. Flue Gas per min.	Flue Gas Properties: Lb. Water Vapor per min.	% by Vol. CO_2	% by Vol. O_2	% by Wt. CO_2	% by Wt. O_2	% by Wt. N_2	% by Wt. H_2O	Specific Vol. cu.ft. per lb.	Total cu. ft. Gas per min.	Gas Vel. ft. per sec.	Water Velocity ft. per sec.
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
G	22	1.941	28 890	33 640	1155	922	6.635	3.661	11.28	0.580	4.2	14.7	4.20	14.70	6.03	15.30	42.18	476	11.1	1.58
	23	1.901	28 290	36 830	1151	851	7.250	4.100	11.56	0.560							42.08	487	11.3	1.85
	24	1.858	27 610	36 680	1140	795	7.663	4.370	11.66	0.562							41.80	488	11.4	2.09
	25	1.812	26 950	33 950	1133	769	7.501	4.182	11.55	0.560							41.61	481	11.2	1.81
H	26	0.798	11 870	18 210	906	666	4.271	2.590	5.77	0.262	3.6	15.9	5.19	16.66	73.95	4.49	35.55	205	4.77	0.89
	27	0.728	10 830	19 070	910	611	4.640	2.955	5.73	0.253							35.68	204	4.75	1.21
I	28	0.775	11 530	13 580	855	618	3.851	2.081	12.79	0.448	2.4	17.4	3.50	18.50	74.70	3.40	34.10	436	10.2	0.95
	29	0.693	10 310	22 370	852	537	5.762	3.942	13.01	0.445							34.00	443	10.3	1.51
J	30	3.464	51 550	56 270	1429	1141	8.960	4.675	12.26	1.025	5.1	12.6	7.70	12.60	71.70	8.40	55.37	508	11.8	1.20
	31	3.396	50 520	55 150	1412	1071	9.350	4.880	12.02	0.821			Estim	ated			40.3	485	11.3	2.17
	32	3.619	53 800	52 500	1424	1059	9.520	4.700	12.28	0.808			Estim	ated			40.0	490	11.4	2.12
K	33	2.220	30 101	54 220	1236	978	8.460	5.260	21.46	1.06			Estim	ated			44.7	960	22.3	0.88
	34	2.525	37 570	49 300	1232	927	8.880	5.040	21.21	1.01			Estim	ated			44.7	948	22.1	1.09
	35	2.122	31 600	51 540	1219	872	9.040	5.600	21.23	1.023			Estim	ated			43.9	932	21.7	1.73
	36	2.068	30 740	53 350	1220	857	9.300	5.900	21.27	1.014			Estim	ated			43.9	934	21.7	1.95
L	37	5.623	83 720	73 200	1608	1369	10.860	5.071	12.71	1.100	8.4	8.6	11.46	8.56	72.10	8.55	55.36	704	16.4	0.46
	38	6.019	89 500	83 790	1645	1334	12.320	5.960	12.58	1.070							56.35	706	16.4	1.71
	39	5.624	83 800	81 330	1636	1295	12.100	5.960	12.94	1.120							56.06	726	16.9	1.68
	40	5.653	84 100	84 860	1613	1251	12.790	6.430	12.64	1.016							55.50	702	16.3	1.74

TABLE 3 (CONCLUDED)
CALCULATIONS AND RESULTS

Series	Test No.	$Av. \left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right]$	Heat by Radiation B.t.u. per hr. H_r	Heat by Convection B.t.u. per hr. H_c	Average Flue Gas Temperature deg. F.	Log. Mean Temp. Diff. T	Total Coef. of Heat Trans. B.t.u. sq. ft. deg. hr. K_t	Convection Coef. of Heat Trans. B.t.u. sq. ft. deg. hr. K_c	Lb. Flue Gas per min.	Flue Gas Properties								Total cu.ft. Gas per min.	Gas Vel. ft. per sec.	Water Velocity ft. per sec.
										Lb. Water Vapor per min.	% by Vol. CO_2	% by Vol. O_2	% by Wt. CO_2	% by Wt. O_2	% by Wt. N_2	% by Wt. H_2O	Specific Vol. cu.ft. per lb.			
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
M	41	4.543	67 600	59 710	1494	1237	9.760	4.575	5.48	0.843	11.4	2.3	14.55	2.13	70.00	15.35	54.25	298	6.92	1.59
	42	5.940	88 400	76 360	1618	1299	12.020	5.575	6.37	1.010	13.5	1.9	17.24	1.77	68.80	15.41	57.82	368	8.58	1.18
	43	4.920	73 200	76 050	1567	1219	11.610	5.920	5.98	0.925							56.42	338	7.86	2.40
	44	5.925	88 200	70 250	1606	1230	12.200	5.420	5.81	0.909							57.52	340	7.91	2.37
N	45	4.118	61 260	80 740	1476	1232	10.920	6.210	19.63	1.37	6.6	11.7	9.30	12.00	72.20	6.90	51.08	1003	23.3	2.32
	46	3.292	49 000	75 400	1303	1100	10.720	6.500	19.75	1.34							46.50	918	21.4	2.22
	47	4.396	65 400	79 870	1478	1132	12.150	6.681	19.48	1.34							51.12	996	23.2	2.27
	48	4.820	71 750	75 740	1511	1146	12.180	6.261	19.73	1.40							52.00	1026	23.9	2.23
O	49	1.584	23 550	41 260	1053	801	7.670	4.885	20.69	0.861	3.4	16.1	4.93	16.94	74.10	4.95	40.08	830	19.3	1.04
	50	1.431	21 300	41 140	1059	751	7.890	5.195	20.07	0.871							40.23	808	18.8	1.31
	51	1.484	22 080	41 140	1062	716	8.361	5.440	20.69	0.864							40.32	838	19.5	1.66
	52	1.512	22 500	36 370	1051	689	8.100	5.000	20.35	0.830							40.03	813	18.9	1.74
P	53	3.062	45 600	61 200	1357	1091	9.280	5.318	15.06	0.951	6.4	11.9	8.96	12.11	72.70	6.74	47.64	718	16.7	2.02
	54	4.040	60 120	72 540	1506	1194	10.540	5.760	15.34	1.093							51.58	792	18.4	1.67
	55	3.826	56 920	79 910	1524	1175	11.030	6.445	15.24	1.055							52.10	794	16.5	2.29
	56	4.211	62 700	73 250	1518	1151	11.180	6.035	15.58	1.011							51.82	808	18.8	2.32
Q	57	6.272	93 300	90 140	1688	1430	12.150	5.980	15.65	1.322	8.7	8.4	11.85	8.32	72.00	8.57	57.95	907	21.1	2.54
	58	5.198	77 350	101 860	1702	1350	12.580	7.150	15.01	1.302							58.38	876	20.4	2.53
R	59	4.775	71 100	77 040	1555	1293	10.860	5.650	12.29	1.067	8.0	9.7	10.90	9.63	71.60	8.75	53.39	656	15.3	2.40
	60	5.652	84 100	87 650	1655	1300	12.530	6.400	12.38	1.091							56.02	694	16.2	2.49
S	61	1.572	23 380	47 270	1137	906	7.390	4.940	15.88	0.898	4.3	14.5	6.16	15.10	73.90	5.12	41.70	662	15.4	1.57
	62	1.647	24 500	47 330	1130	828	8.220	5.418	15.65	0.790							41.50	650	15.1	1.82
	63	1.690	25 130	43 090	1125	790	8.180	5.167	15.54	0.775							41.38	644	15.0	1.80
	64	1.757	26 120	39 570	1133	773	8.050	4.850	15.50	0.759							41.84	649	15.1	1.73

Illinois over a period of a year and was as follows (in per cent by volume):

CO_2	O_2	C_2H_4	C_6H_6	H_2	CO	CH_4	C_2H_6	N_2
7.0	0.6	11.2	0.4	38.0	22.0	4.0	3.0	13.8

From this analysis calculations show that 0.777 lb. of water vapor are formed per lb. of gas burned. The water vapor in the secondary air was determined from the relative humidity of the room air. The water vapor in the primary air was determined by assuming a saturated condition at a 40-lb. gage pressure (supply pressure) and then assuming adiabatic expansion to the burner pressure. The sum of these three water contents gave the total weight of water vapor appearing in the flue gas. This is tabulated in col. 9, Table 3.

From the Orsat analysis and the water content, the analysis by weight of the flue gas was calculated, and tabulated in cols. 12 to 15 inclusive. From the weight analysis, the gas constants as given by Goodenough,* and the "perfect gas law," the specific volume for each test was calculated and recorded in col. 16 at the temperature of col. 4.

From the total weight of gas supplied (the sum of the primary air, secondary air, and illuminating gas, indicated in col. 8) the total volume of flue gas flowing per minute was determined by multiplying this total weight by the specific volume. This total volume is given in col. 17.

The gas velocity at the temperature of col. 4 was determined by dividing the volume of flue gas in cu. ft. per sec. by the effective gas passage area which was 0.716 sq. ft. This velocity is the average gas velocity (col. 18).

V. Discussion of Results

29. *Coefficients of Heat Transmission.*—The coefficients K_t and K_c were plotted against the logarithmic mean temperature difference both for gas flow rates $\left(\frac{W}{A}\right)$ of 8.4, 16.8, 22.3, 29.6 lb. of gas per sq. ft. of gas passage area per min., and for gas velocities of 6, 9, 11, 15, and 20 ft. per sec. The relations between the coefficients and the mass velocity are shown in Figs. 9, 10, and 11. The relations between the volume velocities are shown in Figs. 12, 13, and 14.

For each point indicated the test number, steam saturation temperature, and flue-gas temperature are shown. From the graphs it is evident that there are four variables which affect these coefficients: namely, (a) the rate of gas flow; (b) the mean temperature difference; (c) the temperature of the flue gas; (d) the boiler pressure.

*"Principles of Thermodynamics;" p. 271; Holt and Co.

The reasons for these variations are as follows: (a) According to the film theory of heat transmission the higher the gas velocity becomes, the more violent becomes the scrubbing action of the flowing gases upon the motionless gas film in intimate contact with the heating surface, thus decreasing the thickness of gas film through which heat must pass by conduction. (b) As the mean temperature difference increases, the thermal head between the flue gas and the water increases, thus increasing the rate of heat transmission. (c) As the temperature of the flue gas increases, the thermal conductivity, specific heat, and radiating power of the gas increase, all tending to increase the rate of heat transmission. (d) As the boiler pressure is increased the water velocity increases which decreases the thickness of the water film on the inside of the tube, and the density, specific heat, and conductivity of the steam bubbles, and the specific heat and conductivity of the water increase, all tending to increase the rate of heat transmission.

Figures 11 and 14 show average values of K_t and K_c plotted against the logarithmic mean temperature difference. These average values were assumed to be on the 300-deg. saturation temperature lines.

The K_t results are not consistent, as would be expected, on account of the varying effect of radiation from the fire-clay flue.

The K_c results are consistent on the $\frac{W}{A}$ basis but not on the gas velocity basis at the lower velocities. This is probably because of the fact that for any test the gas velocity varies from a maximum at the hot end of the tube to a minimum at the cold end, but not with a straight line relationship, due to the temperature gradient of the gas along the flue; this velocity variation will be greater at low velocities than at high because of the steeper temperature gradient along the flue at low velocities.

The following empirical formulas were determined from the results:

$$K_c = (0.000977\ \Delta T + 0.0025\ T_w + 0.0855\ W_a + 0.00294\ T_g - 1.83) \tag{11}$$

$$K_c = (0.000715\ \Delta T + 0.00153\ T_w + 0.0927\ V_s + 0.00282\ T_s - 0.9) \tag{12}$$

where

K_c = "convection coefficient" of heat transmission in B.t.u. per sq. ft. per deg. F. per hr.

ΔT = log. mean temp. diff. between flue gas and water in tube.

T_w = temperature of water in tube in deg. F.

W_a = rate of gas flow $\left(\frac{W}{A}\right)$ in lb. per min. per sq. ft. of gas passage area.

V_s = velocity of gas in ft. per sec.

T_g = average temp. of gas in deg. F.

Equation (11) has the greater accuracy because of the greater consistency of the coefficient relations with respect to rate of gas flow in lb. per sq. ft. per min. than with respect to the gas velocity in ft. per sec. No attempt was made to express K_t relations in the form of an equation because of the complications introduced by radiation from the flue.

K_c represents a convection coefficient obtained with water velocities shown in col. 2, Table 4. These velocities were caused by a combination of both convection and radiation and hence would be greater than the velocities caused by convected heat energy alone.*

30. *Relations Between Coefficient of Heat Transmission and Water Velocity.*—It was difficult to determine the effect of the water velocity on the coefficients of heat transmission because of the fact that for each test of a series the steam pressure was changed with a resulting change in water velocity and coefficients. To separate the effects of changes in steam pressure and water velocity, the tests were plotted on logarithmic paper separately for each test pressure with K_t, water velocity, and temperature difference between tube and water as coordinates. The relations of the velocities with the variables just given were irregular but when broadly interpreted seemed to follow the expression

$$K_t = AV_m^{0.128} \qquad (13)$$

where

A = a constant for any one steam pressure.

V_m = water velocity in tube in ft. per min.

Thus doubling the water velocity increases the coefficient K_t only about 9.5 per cent.

31. *Circulation Phenomena.*—When the water in the apparatus is first heated, a convection velocity results because of the density difference between the hot water in the "up-header" and the cold water in the "down-header." This convection velocity tends to decrease as

*K_c may be corrected by Equation (13) and Figs. 15 and 16. For example, in Test 23 the tube velocity was 2.14 ft. per sec; H_t was 65120; H_c was 36830; and K_c was 4.1. Then for pure convection heat transfer the water velocity from Fig. 16 would be given at $\frac{36830}{10.55}$ or 3490 B.t.u. per sq. ft. per hr. or 1.4 ft. per sec. and the coefficient correction factor would be $\left(\frac{1.4 \times 60}{2.14 \times 60}\right)^{0.128}$ or 0.95 and the true K_c would be $4.1 \times 0.95 = 3.9$.

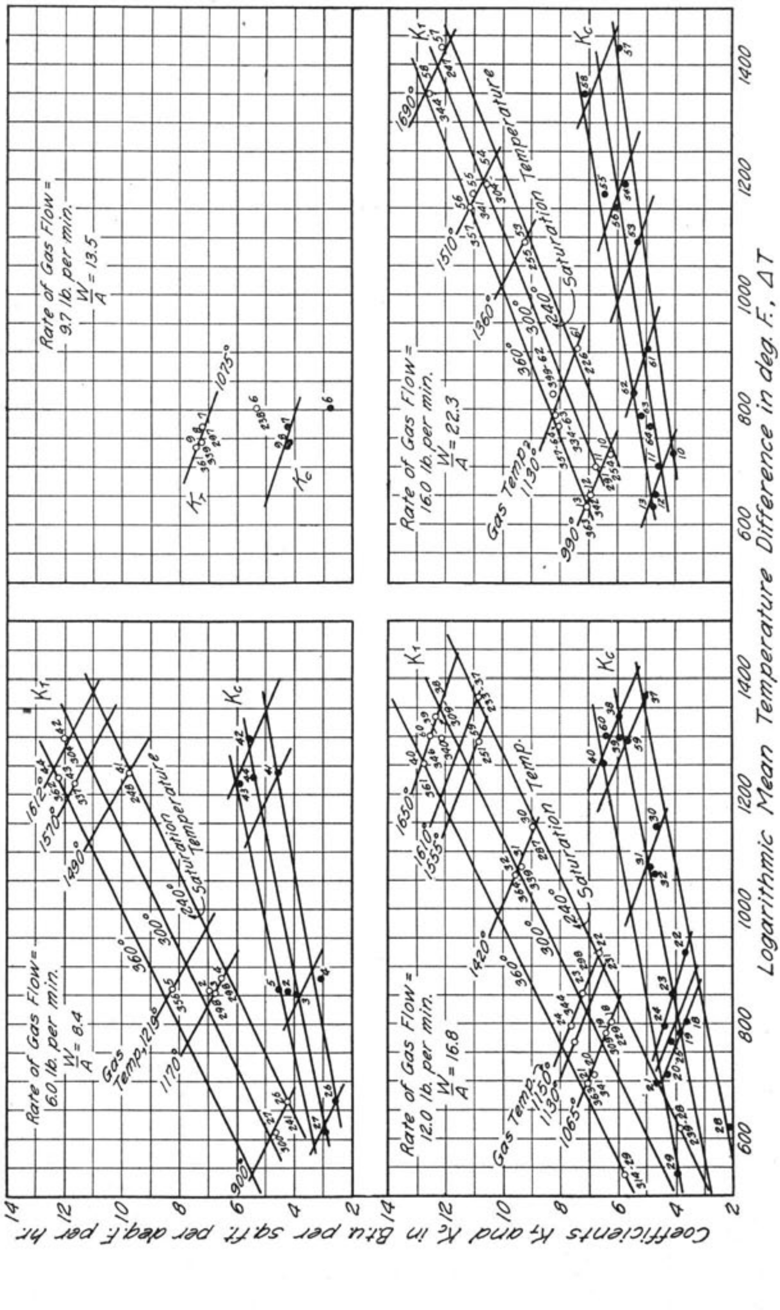

FIG. 9. RELATIONS BETWEEN K_t, K_c, AND TEMPERATURE DIFFERENCE

$\left(\frac{W}{A} = 8.4, 13.5, 16.8, \text{ and } 22.3\right)$

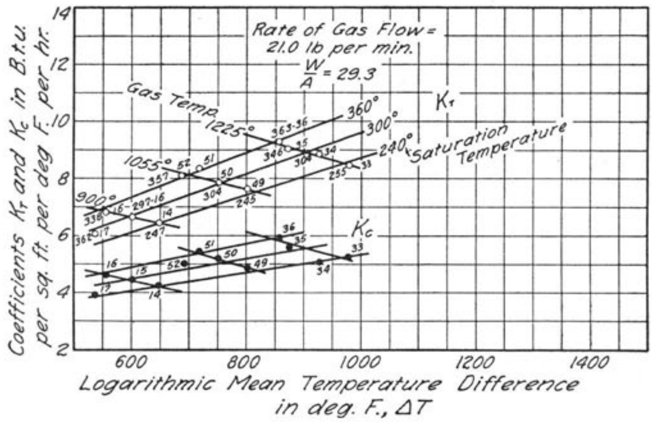

FIG. 10. RELATIONS BETWEEN K , K_c, AND TEMPERATURE DIFFERENCE $\left(\frac{W}{A} = 29.3\right)$

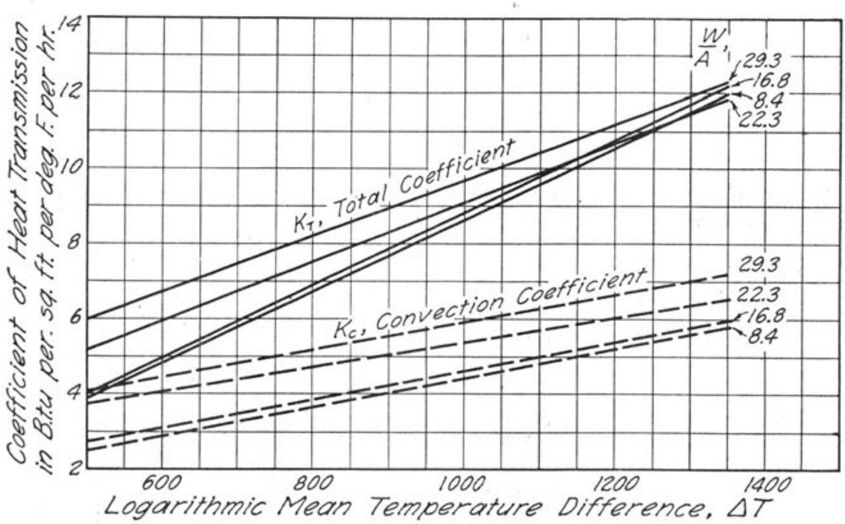

FIG. 11. RELATIONS BETWEEN AVERAGE VALUES OF K_t, K_c, AND TEMPERATURE DIFFERENCE $\left(\frac{W}{A}\text{ Basis}\right)$

TABLE 4

WATER VELOCITY DATA

Series	Test No.	Equivalent Steaming Rate lb. per hr.	Tube Entry Water Vel. ft. per sec.	Measured Temp. Rise Between Couples 7w and 1w deg. F.	Calculated Temp. Rise Between Couples 7w and 1w deg. F.	Wt. Water Entering Tube lb. per hr.	B.t.u. to Water per hr.	B.t.u. to Steam per hr.	Cu. Ft. Steam per hr.	Tube Steam Vel. ft. per sec.	Mixture Vel. Tube Outlet ft. per sec.	Av. Tube Mix. Vel. ft. per sec.
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11
A	2	64.2	1.27	2.86	3.0	20 500	61 920				1.27	
	*3	40.5	1.67	0.82	1.6	26 400	43 300	15 450	86	0.31	1.98	1.83
B	4	81.2	1.22	1.92	1.6	19 250	31 600	26 760	141	0.51	1.73	1.48
	5	84.0	1.38	1.19	0.7	19 750	14 490	57 620	144	0.52	1.90	1.64
C	6	45.1	0.67	5.44	2.6	11 010	28 760	15 280	204	0.74	1.41	1.04
	7	61.8	1.26	2.53	1.6	19 840	32 510	23 980	130	0.47	1.73	1.50
	*8	63.6	1.63	1.39	0.9	25 050	23 500	30 600	94	0.34	1.97	1.90
	9	67.3	1.54	1.39	0.7	23 410	17 220	37 540	87	0.32	1.86	1.70
D	10	47.3	1.00	1.70	2.8	16 230	45 900				1.00	
	11	57.2	1.27	0.94	1.5	20 020	31 500	16 450	95	0.35	1.62	1.45
	*12	54.3	1.55		0.9	23 820	22 380	22 750	67	0.25	1.80	1.68
	*13	49.1	1.78		0.7	26 910	19 800	24 400	56	0.20	1.98	1.88
E	*14	47.1	1.43	3.70	3.0	23 130	69 700				1.43	
	*15	40.2	1.61	2.78	1.5	24 800	38 200	1 980	10	0.038	1.64	1.62
	16	44.6	2.12	1.67	0.9	32 610	30 600	6 730	20	0.074	2.19	2.15
	17	36.2	2.30	1.64	0.7	34 700	25 600	6 040	13	0.049	2.35	2.30
F	*18	58.2	1.72	4.65	4.1	27 910	111 500				1.72	
	*19	42.8	1.97	1.51	1.3	30 860	41 410	19 060	86	0.32	2.29	2.13
	20	54.5	2.02	1.14	0.9	31 060	29 180	1 939	55	0.20	2.22	2.12
	*21	58.8	1.94	0.86	0.7	29 300	21 600	27 470	60	0.22	2.16	2.05

TABLE 4 (CONTINUED)

WATER VELOCITY DATA

Series	Test No.	Equivalent Steaming Rate lb. per hr.	Tube Entry Water Vel. ft. per sec.	Measured Temp. Rise Between Couples 7w and 1w deg. F.	Calculated Temp. Rise Between Couples 7w and 1w deg. F.	Wt. Water Entering Tube lb. per hr.	B.t.u. to Water per hr.	B.t.u. to Steam per hr.	Cu. Ft. Steam per hr.	Tube Steam Vel. ft. per sec.	Mixture Vel. Tube Outlet ft. per sec.	Av. Tube Mix. Vel. ft. per sec.
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
G	*22	66.7	1.83	3.36	4.0	29 860	119 800				1.83	
	*23	73.3	2.14	1.59	1.5	33 700	51 920	11 230	59	0.22	2.36	2.25
	24	72.7	2.42	0.21	0.9	37 100	34 880	26 850	76	0.28	2.70	2.56
	*25	74.0	2.10	0.21	0.7	31 800	23 430	34 670	78	0.29	2.39	2.25
H	*26	24.2	1.03	2.28	3.5	16 770	59 000				1.03	
	*27	31.6	1.41	0.69	1.5	22 200	34 190				1.41	
I	*28	28.6	1.11	2.36	3.7	18 000	66 990				1.11	
	*29	33.9	1.75	0.86	1.3	27 290	36 630				1.75	
J	30	100	1.40	1.56	1.7	22 150	38 450	67 530	419	1.53	2.93	2.16
	*31	121	2.51		0.9	38 600	36 180	66 990	206	0.75	3.26	2.88
	*32	120	2.46		0.7	37 200	27 430	76 110	172	0.63	3.09	2.77
K	33	84.3	1.02		2.8	16 370	46 300	39 390	398	1.45	2.47	1.79
	34	91.9	1.26		1.5	19 750	30 520	54 260	271	0.99	2.25	1.76
	*35	89.9	2.01		0.9	30 750	28 910	51 600	147	0.54	2.55	2.28
	*36	98.3	2.25		0.7	34 200	25 200	56 110	129	0.47	2.72	2.53
L	37	163	0.53		3.9	8 590	33 620	121 930	1690	6.17	6.70	3.61
	38	184	1.98		1.3	31 050	41 690	129 300	588	2.14	4.12	3.05
	39	183	1.94		0.9	30 310	28 480	134 050	419	1.53	3.47	2.71
	40	194	2.02		0.7	30 550	22 500	143 690	328	1.19	3.21	2.57
M	41	129	1.85		3.2	29 820	96 100	29 670	297	1.08	2.93	2.39
	42	177	1.37		1.5	21 510	33 290	129 310	469	1.71	3.08	2.22
	*43	161	2.78		0.9	42 800	40 210	107 470	262	0.96	3.74	3.26
	*44	183	2.74		0.7	41 390	30 500	125 070	227	0.83	3.57	3.15

TABLE 4 (CONCLUDED)

WATER VELOCITY DATA

Series	Test No.	Equivalent Steaming Rate lb. per hr.	Tube Entry Water Vel. ft. per sec.	Measured Temp. Rise Between Couples 7w and 1w deg. F.	Calculated Temp. Rise Between Couples 7w and 1w deg. F.	Wt. Water Entering Tube lb. per hr.	B.t.u. to Water per hr.	B.t.u. to Steam per hr.	Cu. Ft. Steam per hr.	Tube Steam Vel. ft. per sec.	Mixture Vel. Tube Outlet ft. per sec.	Av. Tube Mix. Vel. ft. per sec.
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
N	*45	150	2.69		3.8	43 600	166 700				2.69	
	*46	135	2.58		1.5	40 500	62 500	60 060	300	1.09	3.67	3.13
	*47	161	2.59		0.9	39 650	37 280	105 610	293	1.07	3.66	3.12
	*48	165	2.63		0.7	39 850	29 310	115 580	246	0.90	3.53	3.08
O	49	60.0	1.20		3.2	19 440	62 700	499	6	0.02	1.22	1.21
	50	64.3	1.51		1.5	23 710	36 670	23 470	117	0.43	1.94	1.72
	*51	68.4	1.92		0.9	29 440	27 640	32 850	92	0.33	2.25	2.08
	*52	69.4	2.01		0.7	30 510	22 440	33 540	77	0.28	2.29	2.15
P	*53	111	2.34		2.2	37 690	83 810	21 440	186	0.68	3.02	2.68
	54	130	1.94		1.5	30 350	47 000	83 540	363	1.33	3.27	2.60
	*55	152	2.65		0.9	40 600	38 210	96 060	260	0.95	3.60	3.13
	*56	159	2.69		0.7	40 700	29 990	103 220	222	0.81	3.50	3.09
Q	*57	192	2.94		3.2	47 400	153 100	28 890	275	1.00	3.94	3.44
	*58	203	2.93		0.7	44 700	32 740	143 802	369	1.35	4.28	3.60
R	*59	153	2.78		1.0	44 720	45 190	101 325	930	3.39	6.17	4.47
	*60	192	2.89		0.7	44 040	32 340	136 625	330	1.22	4.11	3.50
S	*61	73.4	1.82		4.4	29 710	131 000				1.82	
	*62	77.7	2.11		1.5	33 140	51 080	18 680	96	0.35	2.46	2.28
	*63	74.5	2.08		1.0	32 200	33 500	32 281	105	0.38	2.46	2.27
	*64	75.0	2.00		0.7	30 390	22 340	40 610	95	0.35	2.35	2.17

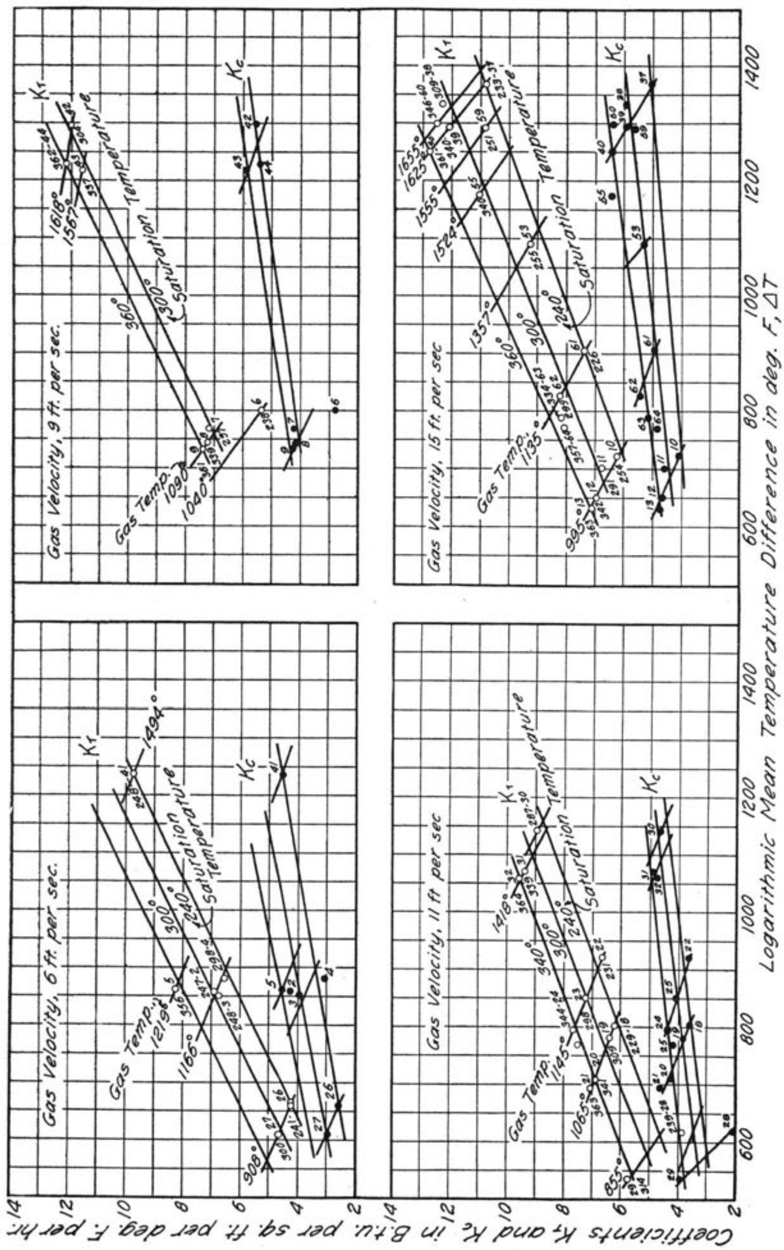

FIG. 12. RELATIONS BETWEEN K_t, K_c, AND TEMPERATURE DIFFERENCE
($V = 6, 9, 11,$ and 15)

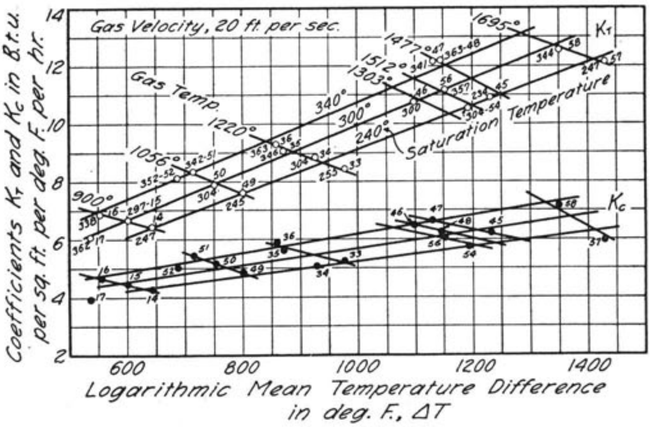

FIG. 13. RELATIONS BETWEEN K_t, K_c, AND TEMPERATURE DIFFERENCE ($V = 20$)

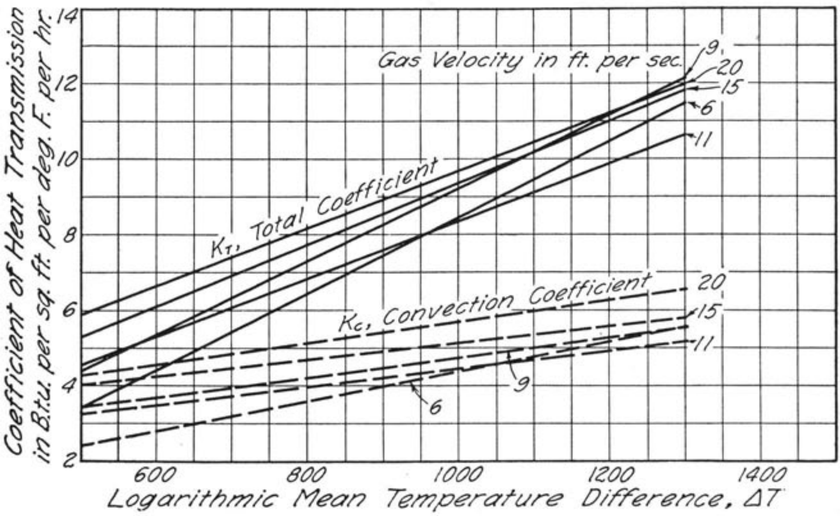

FIG. 14. RELATIONS BETWEEN AVERAGE VALUES OF K_t, K_c, AND TEMPERATURE DIFFERENCE (VELOCITY BASIS)

the water temperature in the apparatus becomes higher. This is because the difference in density between the two water legs becomes less as the temperature increases on account of the fact that the rate of decrease of water density decreases as the temperature increases and therefore the difference in weight between the hot and cold legs for a 2-deg. temperature difference at 400 deg. is less than for a 2-deg. temperature difference at 200 deg.

In addition to the convection effect, at a certain temperature condition in the heating of the water the "bubble" effect begins. This phenomenon is demonstrated in the "air-lift" pump and consists in the decreasing of the density of the fluid in the up-header by the formation of steam bubbles.

This bubble effect is caused by two distinct operations: (1) Part of the water heated to the saturation temperature at the bottom of the up-header may flash into steam as it rises in the up-header, due to the decrease in static pressure of the water. (2) The water in the tube may be heated at a greater rate than can be absorbed by the water alone and hence steam bubbles may be formed in the tube itself.

Consistent correlation of the water velocity with some other variable was difficult. This was due in part to experimental error and possibly also because of some inherent variation in the circulation phenomena. In tests 7, 9, 11, 30, 37, 38, 39, 40, and 41, the manometer action was markedly sluggish and the leads were blown out during the test, which would exclude these tests from consideration.

In the operation of the apparatus, a distinct drop in the manometer reading was noticed when the discharge valve was shut in raising the pressure for the next test. Conversely, a sudden increase in the manometer reading was apparent when the discharge valve was opened with a resulting decrease in drum pressure. It may be then, that the adjusting of the discharge pressure valve during a test accounts for the discrepancy in some of the tests.

The tests starred in Table 4 have been selected* as giving the most representative and probably the most correct water velocities. These are plotted in Figs. 15 and 16.

*The method of selection was to plot separately the water velocities in four separate groups according to the test steam pressure against the "equivalent" steaming rate. It was noted that for each pressure group, a majority of the velocity points defined a fairly regular graph. It was also noticed that the curve so defined was practically identical for all four pressure groups. The test points defining this curve, then, were taken as the more accurate determinations and are those starred in Table 4 and plotted in Figs. 15 and 16. The water velocity determinations were more consistent for the high pressure tests than for the low; this was probably due to the fact that the rate of change (with pressure change) of specific volume and heat content of steam per unit volume is greater at the low pressures than at the higher, and therefore any inconstancy of test pressure would affect the low pressure velocities more than those at the higher pressures.

It was more difficult to keep the pressure constant at the lower ranges than at the higher; this fact also tends to scatter the low pressure velocity points.

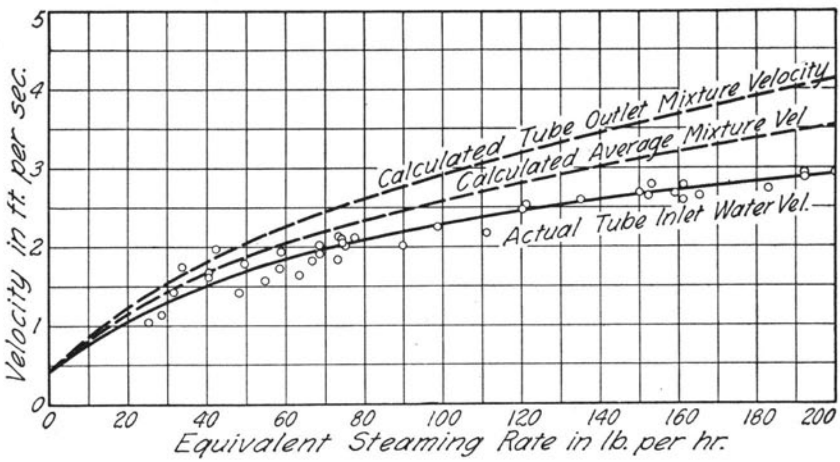

FIG. 15. RELATIONS BETWEEN WATER VELOCITY AND EQUIVALENT STEAMING RATE

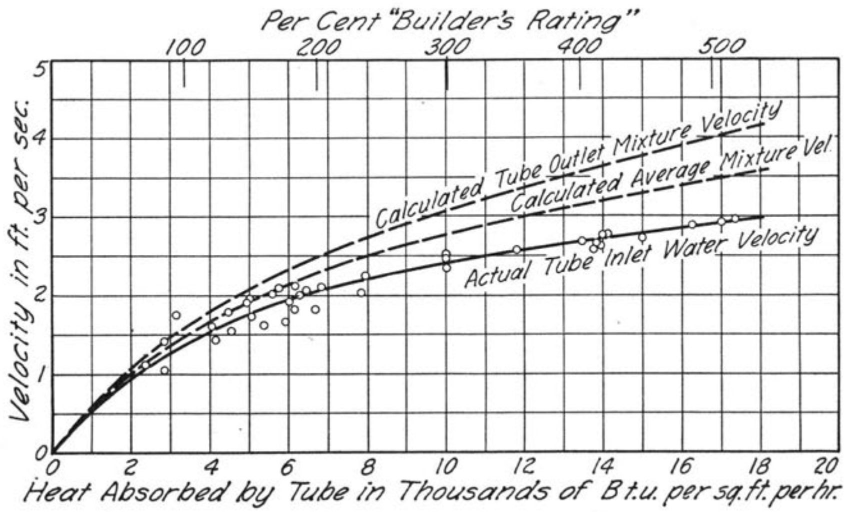

FIG. 16. RELATIONS BETWEEN WATER VELOCITY AND HEAT ABSORBED BY TUBE

Figure 15 shows the variation of the tube inlet water velocity in ft. per sec. with the equivalent steaming rate.* The tube inlet velocity is the down-comer velocity (col. 19, Table 3) multiplied by the ratio of the cross-sectional area of the down-comer to that of the tube, or $\frac{0.0884}{0.0762}$. This graph has been drawn to intersect the Y axis at 0.4 as this is about the average convection velocity for each test series by calculation.

As the water moves along the tube due to circulation it receives heat from the tube wall with a resulting rise in temperature until the saturation temperature is reached; after this additional heat energy received is converted into the latent heat of the water and thus at the tube outlet a mixture of steam and water is discharged.

When the water velocity entering the tube and the water temperature rise from one end of the tube to the other is known, the amount of heat energy appearing in the water per unit of time can be calculated; if this amount is subtracted from the total heat energy received by the tube, the remainder will be the energy appearing in steam bubbles (latent heat of vaporization). When the pressure at the tube discharge point is known, the volume of the steam and water mixture leaving the tube can be calculated.

The measured increase in temperatures between 1W and 7W is indicated in col 3, Table 4. Unfortunately the temperature indications for these couples were not sufficiently accurate to permit using them in these calculations, as an error of 1 deg. F. in temperature measurement involves an error of 25 per cent or more in this calculation, since the maximum temperature rise was about 4 deg. This was probably due to a break of insulation on account of the high gas temperatures in test 30.

The maximum temperature rise between couples 7W and 1W was calculated (col. 4) by making the following assumptions: (1) that the water leaving the drum and entering the down-comer was at the drum saturation temperature; (2) that the temperature drop in the down-comer was negligible;† (3) that the water reached the saturation temperature at couple 1W; and (4) that the water level in the drum was constant at the center-line.

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*The equivalent steaming rate, col. 1, Table 4, is the sum of the steam actually generated in lb. per hr. plus the equivalent weight of steam necessary to supply the radiation (or insulation) losses by condensing; the latter being equal to the insulation loss in B.t.u. per hr. from Fig. 6, divided by the latent heat of vaporization at drum temperature.

†The radiating surface of the down-comer was 11.8 sq. ft., and the total energy loss per hour was calculated from Fig. 6; this loss divided by the weight of water circulated per hr. (col. 5) would give the temperature drop of the water passing through the down-comer. This calculated drop in the down-comer averaged about 0.1 deg. F.

The temperature of the water leaving the tube was found by determining the saturation pressure (and temperature) at that point; the drum steam pressure was known and the additional static water pressure at the top end of the tube was calculated from the difference in elevation between that point and the drum center-line. The total pressure then at couple 1W was known and the saturation temperature taken from steam tables.

By referring to cols. 3 and 4 it is seen that the calculated temperature rise compares favorably with the measured temperature rise.

This calculated temperature rise was then used to calculate the volume of steam passing the outlet end of the test tube from the expression

$$V_s = \frac{H_t - \Delta T \times C \times W}{H_f} \tag{14}$$

where

V_s = cu. ft. of steam passing through outlet of test tube per hr. (col. 8)

H_t = total heat energy received by test tube in B.t.u. per hr. (col. 18, Table 1)

ΔT = calculated temp. rise of water in passing through test tube

W = weight of water in lb. per hr. passing through tube (col. 5)

C = specific heat of water at test temperature

H_f = heating value per cu. ft. of steam at tube outlet pressure

Dividing V_s by the cross-sectional area of the tube gives the steam velocity through the outlet end of the tube; this is expressed in ft. per sec. in col 9. The summation of the steam velocity and the water velocity at the tube outlet has been called the "mixture" velocity and is shown in col. 10.

Assuming that half of the steam formation has been completed when the water has traversed half the tube length, an "average mixture" velocity has been calculated (col. 11) by adding one half the steam velocity (col. 9) to the water velocity (col. 2). The relation between the water and steam velocities and the equivalent steaming rate is shown in Fig. 15. This graph has been drawn intentionally to indicate approximately a water velocity of 0.4 ft. per sec. at zero evaporation, because this is about the average calculated convection velocity.*

Figure 16 indicates the relation obtained from col. 18, Table 1, between the water and mixture velocities and the heat energy re-

*For calculation of convection velocity see: Harding and Willard, "Mechanical Equipment of Buildings," Vol. I, Chap. X.

ceived by the test tube. The points indicated are actual water velocity determinations from the selected group of tests. The graph was drawn through the origin because with no heat addition there could be no motive force for the water, and hence the velocity would be zero. The calculated points for the mixture velocities are not shown because of the assumptions made in the calculations. The "Per Cent Builder's Rating" indicated is allowing 3352 B.t.u. per sq. ft. per hr.

The relation between the entering velocity of the water and the rate of energy addition for the range of the tests can be expressed

$$V_w = 0.0278\ H^{0.484} \tag{15}$$

where

V = velocity of water entering tube in ft. per sec.

H = total energy added to tube in B.t.u. per sq. ft. of heating surface per hr.

The mixture velocity leaving the tube can be expressed

$$V_m = 0.0146\ H^{0.58} \tag{16}$$

where

V_m = velocity of steam and water mixture leaving tube in ft. per sec.

From the graphs it is evident that the velocity of the water entering the tube tends to reach a practical maximum at slightly over three feet a second.* The test tube circulation results should be applied only to the lowest rows of tubes in a boiler.†

In conducting the tests it was noticed that for each series the water velocity increased as the steam pressure was increased for each successive test, and an attempt was made to formulate an expression correlating circulation with the physical characteristics of the steam and water mixture.

The "induced" water velocity (average mixture velocity minus convection water velocity) should vary with the size of the bubbles formed, the difference in density between the steam bubbles and the water, the rate at which the bubbles are formed, and the frictional resistance offered to flow.

Several functions involving these variables were calculated but none were as consistent as the velocity—heat-absorption relation given in Fig. 16. The following formulas were derived as being the most representative of the relations existing between the variables named:

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*The maximum velocity in a 1½-in. "fire-row" tube was found to be 2.92 ft. per sec. See "Water Velocities in a Yarrow Boiler." The Engineer (London), April 9, 1926.

†For a comparative study of the circulation in the various tubes of a boiler, from bottom to top, see "Tests of Marine Boilers," by Kreisinger, Blizard, Mumford, etc. Bur. of Mines Bul. 214.

$$V_i = 0.379\,\theta^{0.286} \tag{17}$$

where

$$\theta = \frac{K_t^2\sqrt{\dfrac{H_f}{R}}}{\nu} \tag{18}$$

V_i = induced water velocity in ft. per sec. (average tube mixture velocity minus convection water velocity)

K_t = "total" coefficient of heat transmission

H_f = B.t.u. per cu. ft. of steam at test pressure

R = ratio of water density to steam density at test temperature and pressure

ν = kinematical viscosity of water at test temperature times 100 000 in ft.² per sec.

32. *Temperature Gradient from Flue Gas to Water.*—The temperature gradient from the flue gas to the water in the test tube varied with (a) the rate of gas flow; (b) the temperature of gas; (c) the steam pressure, or temperature.

The correlations of these variables for all tests are shown in Fig. 17 with the logarithmic mean temperature difference (ΔT) used as a base. Each graph is plotted for a different rate of gas flow $\frac{W}{A}$. The flue-gas temperature is taken from Table 3 as the average gas temperature. The tube temperature was taken from couple 5W.

Referring to Fig. 17, for example, for test 52 the gas temperature was 1051 deg. and the steam temperature was 357 deg. Following the dash line as shown, the difference in the temperature between the outside surface of the tube and the water in the tube was 53 deg. The temperature of the tube, outside surface, was the steam temperature, 357, plus 53, or 410 deg.

Care should be used in applying these temperature gradient data to other apparatus, because of the fact that the gradients for these tests are higher than would exist for pure convection heat transfer on account of the radiating effect of the flue lining.

IV. Conclusions

33. *Conclusions.*—As a result of this investigation the following conclusions may be drawn:

(1) The coefficient of heat transmission of the apparatus is affected by: (a) the rate of gas flow; (b) the temperature difference between flue gas and water; (c) the pressure at which steam is gener-

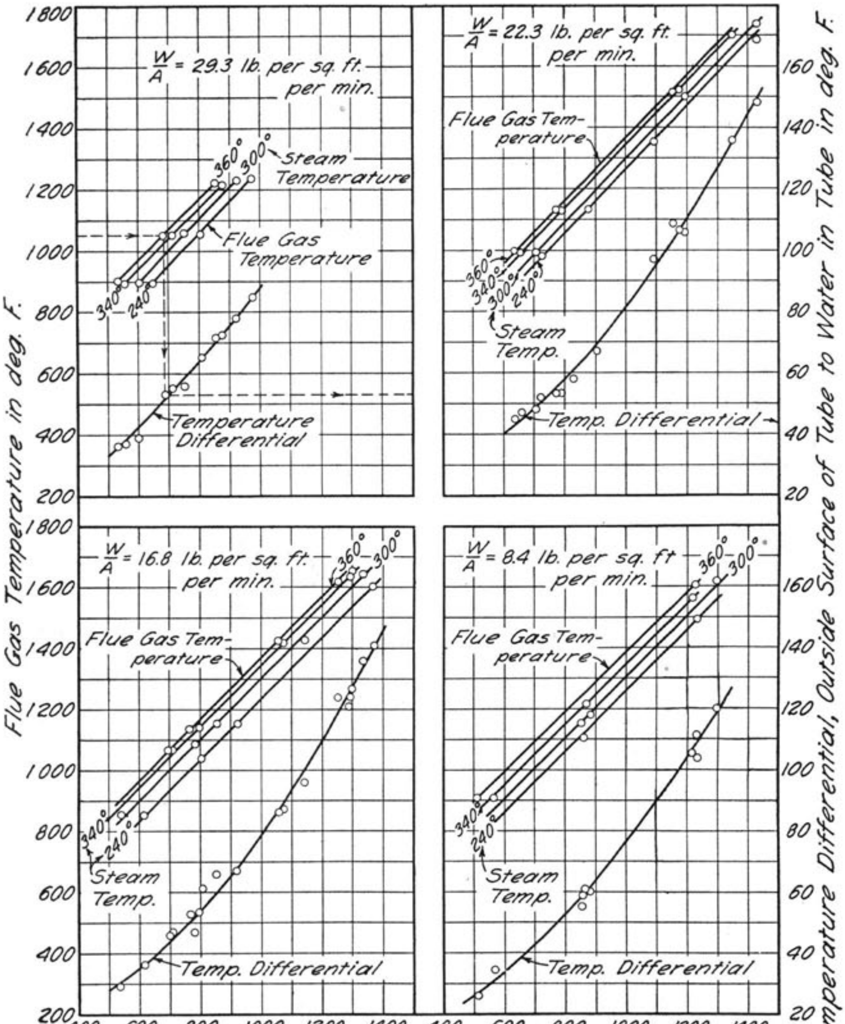

FIG. 17. RELATIONS BETWEEN TEMPERATURE GRADIENT AND TEMPERATURE DIFFERENCE $\left(\frac{W}{A} = 29.3, 22.3, 16.8, \text{ and } 8.4\right)$

ated; (d) the temperature of the gas stream. The coefficients may be expressed in terms of the variables by means of equations (11) and (12), p. 35.

(2) The water velocity in the tube has slight effect on the overall coefficient of heat transmission as expressed broadly by equation (13).

(3) The water velocity in the inclined tube of the apparatus varies as the steam rate and total energy input, and tends to approach a practical maximum at slightly over 3 ft. per sec.

(4) The water velocity is affected by the steam pressure as indicated by equation (17).

(5) The temperature gradient from the flue gas to the water in the tube varies with the flue-gas temperature, the rate of gas flow, and the steam pressure, as shown by Fig. 17.

APPENDIX

34. *Additional Tests with a Tube Coated with Scale.*—With the assistance of R. W. Shields,* a series of sixteen tests were conducted on another tube withdrawn from actual boiler service because of scale troubles.

The tube was furnished by Mr. M. M. Shepard of the Franklin County Coal Company from their steam plant at the Royalton Mine.

This tube when shipped was incrusted with scale of about one-half inch thickness, but upon arrival at the laboratory most of the scale had been loosened.

After the loose scale had been removed and the tube was ready to place in the apparatus for test purposes the scale thickness was carefully estimated to be 0.083 in.

The scale was found to have the following analysis:

	Per cent
Silicon dioxide (SiO_2)	3.80
Iron and aluminum oxide (R_2O_3)	2.22
Calcium oxide (CaO)	36.52
Magnesium oxide (MgO)	3.26
Sulphur Trioxide (SO_3)	50.88
CO_2, moisture and organic matter	3.21
Alkalies (NaO_2) by difference	0.11
	100.00

The external diameter of the tube was 3.93 in. The metal thickness was 0.13 in.

The tests were conducted in the same manner as for the previous "commercially clean" tube.

The arrangement of apparatus and general procedure were the same as for the previous tests with the exception that no water velocities were taken and no water temperatures were obtained on the inside of the tube. Two additional thermocouples were added to the inside surface of the casing for obtaining the average inside casing temperature. This gave a total of four casing temperatures at various distances along the tube for computing the "Heat by Radiation" (H_r, col. 2, Table 3). In the discussion on page 29, the assumption was made that the temperature gradient between the casing and the boiler tube could be represented by a straight line plotted on logarithmic paper. These two additional casing temperatures further justify this

*Mr. Shield's work was done as the laboratory requirement toward the degree of Master of Science in Mechanical Engineering.

TABLE 5

SUMMARY OF RESULTS ON TUBE NO. 2

Length = 10 ft. ¾ in. O.D. = 3.93 in. Metal Thickness = 0.13 in. Scale Thickness = 0.083 in.
Heating Surface = 10.36 sq. ft.

Series	Test No.	Duration hr.	Lb. Steam Condensed-Test	Drum Temperature deg. F.	Total Heat to Steam During Test B.t.u.	Total Heat to Water During Test B.t.u.	Heat Added to Metal During Test B.t.u.	Insulation Loss B.t.u. per hr.	Total Heat Added to Tube B.t.u. per hr.	$Av.\left[\left(\frac{T_1}{1000}\right)^4-\left(\frac{T_2}{1000}\right)^4\right]$	Heat by Radiation B.t.u. per hr.	Heat by Convection B.t.u. per hr.	Average Flue Gas Temperature deg. F.	Log. Mean Temperature Difference	Lb. Flue Gas per min.	K_t	K_c	K'_c
(1)*	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)
T	65	1.25	9.5	245.0	9 106	14 540	3110	10 300	31 700	0.902	13 160	18 540	935	690	4.60	4.44	2.59	
	66	1.25	10.54	288.2	9 750	7 559	1636	15 500	30 650	0.770	11 240	19 410	874	586	5.15	5.05	3.20	
U	67	1.0	60.03	267.8	57 020	19 250	4506	12 950	93 726	2.59	37 810	55 916	1331	1063	11.04	8.50	5.07	5.11
	68	1.0	76.66	302.2	70 200	6 710	1757	16 600	95 267	2.53	36 950	58 317	1364	1062	11.25	8.65	5.30	5.32
	69	1.0	82.95	340.2	73 100	2 533	768	20 150	96 550	2.90	42 350	54 200	1336	996	11.32	9.35	5.25	5.27
V	70	1.0	57.75	255.0	55 080	17 440	4141	11 400	88 060	2.20	32 120	55 940	1257	1002	16.66	8.48	5.38	5.40
	71	1.0	67.50	314.5	61 070	6 300	1636	17 700	86 700	2.48	36 210	50 490	1256	941	16.64	8.89	5.18	5.56
	72	1.0	68.50	350.7	59 810	3 223	929	20 850	84 669	2.00	29 200	55 469	1259	908	16.75	9.00	5.88	5.68
W	73	1.25	10.2	242.5	9 788	17 760	3759	10 400	35 440	0.67	9 780	25 660	819	576	12.01	5.94	4.30	
	74	1.25	20.2	289.4	18 570	1 582	343	15 520	31 890	0.68	9 930	21 960	822	533	12.08	5.77	3.96	
X	75	1.0	69.0	253.2	65 800	12 930	3050	18 850	94 630	3.06	44 700	49 950	1344	1091	5.54	8.38	4.43	4.48
	76	1.0	90.0	308.6	81 800	6 220	1676	18 500	108 196	3.84	56 060	52 130	1414	1105	5.37	9.45	4.55	4.81
	77	1.0	70.75	342.7	62 020	1 176	364	21 400	84 960	2.78	40 600	44 360	1302	959	5.29	8.54	4.46	4.43
Y	78	1.0	114.75	256.1	108 900	7 630	1899	12 850	131 279	3.12	45 560	85 720	1433	1177	18.95	10.75	7.02	6.40
	79	1.0	112.5	303.2	102 200	409	121	17 700	120 430	3.64	53 200	67 260	1398	1095	18.77	10.60	5.93	6.35
	80	1.0	109.25	342.0	95 800	−277	−101	21 200	116 600	3.74	54 600	62 000	1392	1050	19.11	10.72	5.70	6.42

*Columns 1 to 10 are taken from Table 1. Columns 11 to 19 are taken from Table 3.

assumption. The maximum difference between the measured temperature gradient and the assumed temperatures was about one per cent, as indicated from these later tests.

A summary of the test results is given in Table 5. These results were computed by the same methods as were used for the results for the clean tube. The quality of the steam leaving the drum was estimated to be 0.998 for all tests. The water temperature in the tube was assumed to be the same as in the drum where a thermocouple reading was obtained.

In column 18, Table 5, is given the convection heat transmission coefficient K_c for the scaled tube as determined by actual test. In column 19, Table 5, is given the convection heat transmission coefficient K_c' for a clean tube of the same size under the same conditions of gas temperature, gas velocity, and steam pressure. Values of K_c' were calculated from equation (11).

Values of K_c' are not shown for tests 65, 66, 73, and 74 because of the fact that equation (11) does not accurately express the values of the coefficient at low gas temperatures and low gas velocities.

The discrepancy in test 78 is probably due to the fact that the steam pressure was not kept sufficiently constant during the test. The discrepancies in tests 72 and 73 are inexplicable.

In considering the results it should be remembered that approximately 93 per cent of the resistance to heat flow by pure convection from the flue gas to the water in the tube is caused by the motionless gas film on the outside of the tube; about 4 per cent of the resistance occurs in the water film on the inside of the tube; and the remaining 3 per cent of the resistance is caused by the tube wall and the incrusted scale.

The differences then between columns 18 and 19 (Table 5) are caused by an increase of the smallest portion of the overall resistance. These differences are also for heat transmission by convection and conduction of gases only.

If heat radiation is present the reduction in heat transmission due to scale will be greater because of the fact that radiant heat is not absorbed by gases in thin films and therefore the gas film on the outside of the tube offers practically no resistance to radiation.

The differences between columns 18 and 19 (Table 5) therefore give approximately the reduction in heat transmission due to scale 0.083 in. thick for boiler tubes five or six rows above the furnace.

The reduction in heat transfer due to scale where radiant heat is present (approximately comparable to boiler tubes two to four rows of tubes above the furnace) can be obtained by comparing column **17**, Table **5**, with column **6**, Table **3**, for the same rates of gas flow, gas temperature, steam pressure, and absorbed radiation (col. **2**, Table **3**).

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